Mirror Sub-Assembly
End-Effector Design

Becky Butlin

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End-Effector Design

By:
Becky Butlin

MS Plan II – Technical Report for Professor Michael Hill
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**Introduction**

The Optic Assembly Building (OAB) is a facility where large optical mirror units are assembled and installed into Line Replaceable Units (LRUs) for deployment into the National Ignition Facility (NIF) laser system. The New Optics Insertion Device (NOID), shown in Figure 1, is a powered jib crane specially designed to handle large optical assemblies. The NOID arm has three degrees of freedom. It can rotate about the vertical boom, travel up and down the boom, and extend away from and retract in towards the boom. The NOID is used to assist in the assembly of five types of Laser Mirror (LM) LRUs. These five LMs have been creatively named, LM4, LM5, LM6, LM7, and LM8. The LM4 and LM5 LRUs each contain four Mirror Sub-Assemblies (MSAs). The LM6, LM7, and LM8 LRUs each contain 2 MSAs. The MSAs are assembled apart from the LRU and are then installed in the LRU at the LM4-8 workstations shown in Figure 1.

An MSA NOID End-Effector is required to interface with the MSAs and install them into the LRUs. The End-Effector must attach to the Robo-hand on the end of the NOID arm. At the time the MSA NOID End-Effector was being designed the NOID, the LM4-5 workstation, and the LM6-8 workstation were already installed in the OAB. The LRUs and the MSAs designs were also complete. The MSA NOID End-Effector design had to work with the assembly equipment and LRU designs that were already in place.

![Figure 1 LM4-8 Workstation in the OAB](image-url)
**Design Requirements**

The most general, but most important design requirement is that the MSA NOID End-Effector must be able to engage the MSAs and place them in the LM4-8 LRUs in the correct orientation. The MSA design for each LM4-8 LRU is very similar but varies in size. The widths of the MSAs range from 440mm to 685mm. The MSA NOID End-Effector must be able to handle all sizes. All the MSAs have four mounting lugs on the sides as shown in Figure 2. These mounting lugs were added to the MSA design with the intent that they would be used to handle the MSAs in the OAB.

When the MSA is installed in the LRU, there is very little clearance between the MSA and the LRU frame, which creates a space constraint for the End-Effector design. In addition to this space constraint there is also restraint on the length of the End-Effector. The length of the MSA NOID End-Effector has to be less than 495mm or the MSA will interfere with the LRU during installation.

In order to install the MSA into the LRU the MSA NOID End-Effector has to have several degrees of freedom. The End-Effector must be able to manipulate the MSA so that the bolt holes on the back of the MSA can be aligned with the clearance holes in the LRU. The MSA must also be oriented so that it is parallel to the mating surface on the LRU. In order for this to be accomplished the End-Effector has to have a minimum of $\pm 5^\circ$ twist rotation, $\pm 5^\circ$ pitch rotation, and $\pm 20^\circ$ yaw rotation.

The MSA NOID End-Effector must maneuver the MSAs safely and comply with LLNL design standards for lifting devices. To maneuver the MSAs safely, the degrees of freedom of the End-Effector must be controlled to some degree so that the End-Effector is not free to rotate. The End-Effector must be able to hold and maneuver the heaviest MSA which weighs 270 lbs. To comply with the LLNL design standards the static safety factor on each component must be equal to or greater than 3 on yield or 4 on ultimate. Seismic load calculations require a minimum safety factor of 1 on yield or 1.25 on ultimate. There is also a maximum allowable bending moment on the NOID Robo-hand of 6000 in-lbs.

The MSA NOID End-Effector will be used in the OAB, which is a class 100 cleanroom. All exposed materials of the End-Effector must be cleanroom compatible and approved by the NIF cleanliness steering committee. The End-Effector should also be designed in such a way that particle generation is minimized when the End-Effector is used.
The last design requirement is that the final cost of the MSA NOID End-Effector should be minimal. There was no set value, just a requirement to keep the cost down.

**Design Interface Requirements**

As previously discussed, the MSA NOID End-Effector has to attach to the Robo-hand on the NOID arm and interface with each LM4-8 LRU assembly. The End-Effector must install the MSAs into the appropriate LRUs without interfering with the LRU frame. The last piece of equipment the End-Effector must interface with is the MSA Handling Fixture shown in Figure 3. The MSA Handling Fixture is what is used to rotate the MSA from a horizontal to a vertical orientation. The End-Effector must accept the MSA from the MSA Handling Fixture while it is in the vertical orientation.

![Figure 3 MSA Handling Fixture lifting the MSA](image)

**The MSA NOID End-Effector Design**

**Design Overview**

The purpose of the MSA NOID End-Effector is to remove an assembled Mirror Sub-Assembly (MSA) from the MSA Handling Fixture and place it into the LRU. It was a challenge to obtain all the required degrees of freedom in such a compact system. For each degree of freedom, I came up with several conceptual designs. With each conceptual design, I made several sketches
and created a decision matrix to determine which of the design concepts best met the design requirements. I also shared some of the design concepts with the assembly technicians that would be using the End-Effector to gather their thoughts. Once a design concept was chosen, I modeled the design in Pro/DESKTOP and created detailed drawings for each component. The drawings can be found in Appendix A. Great care was taken to make sure the dimensions and tolerances for each part were correct so that the parts would fit together as designed. Once the design was complete a Final Design Review was held. I presented the design to a large group of NIF engineers, mechanical technicians, OAB operations personnel, cleanliness and materials personnel, and safety personnel. The design review was a success and I was given permission to procure the MSA NOID End-Effector.

![Figure 4 MSA NOID End-Effector holding an LM5 MSA](image)

The final End-Effector design is shown in Figure 4. The MSA NOID End-Effector is a mechanical device, which approaches the MSA from the front and holds the MSA using the four mounting lugs on the MSA. A hook and latch mechanism is incorporated into the Side Mounting Plates so that a positive capture of the MSA is assured. The NOID can then lift the MSA from the MSA Handling Fixture and place it into the LRU.

The MSA NOID End-Effector design consists of three Sub-Assemblies as shown in Figure 5. The main function of the Rail Sub-Assembly is to interface with each of the MSAs. The distance between the Side Mounting Plates is adjustable to accommodate the varying widths of the MSAs. The main function of the Tilt Sub-Assembly is to provide Y-axis (yaw) and Z-axis (pitch) rotation. Rotation about the Y-axis is only allowed when a friction nut has been loosened. Rotation about the Z-axis is adjusted manually by turning a hand wheel. The main
function of the Twist Sub-Assembly is to provide rotation about the X-axis (twist). The rotation about the X-axis is also manually adjusted by turning a hand wheel. In this case, turning the hand wheel causes a shaft to rotate, which in turn causes the first two Sub-Assemblies to rotate about the X-axis. The following three sub-sections give more details about the each Sub-Assembly and discuss the conception of the designs.

![Diagram showing three sub-assemblies: Rail Sub-Assembly, Twist Sub-Assembly, and Tilt Sub-Assembly.]

Figure 5 Three Sub-Assemblies of the MSA NOID End-Effector

Rail Sub-Assembly Design
As mentioned earlier, the main function of the Rail Sub-Assembly is to interface with all the MSAs. It has to safely hold the MSAs and protect the optics from particles the End-Effector may generate. The main design constraint of the Rail Sub-Assembly was the size because there is very little clearance between the MSAs and the LRU frames.
There were two main conceptual designs considered for the Rail Sub-Assembly, which will be discussed later in this section. The final design is shown in Figure 6. The main components of the Rail Sub-Assembly are the side mounting plates, the protecting shields, the rail system and ACME rod, and the rail plate.

![Components of the Rail Sub-Assembly](image)

**Figure 6 Components of the Rail Sub-Assembly**
(Additional components shown in figure 7)

Vacuum grippers are often used to handle optical assemblies, but this was not an option for the End-Effector design because of the 4 Mirror Clips on the MSA (shown in Figure 4). The gasket of the vacuum gripper is only allowed to touch the optic outside of the clear aperture (where the NIF laser beam will not hit). In the case of the LM4-8 optics, there is only 14 mm around the edge of the optic that is outside of the clear aperture. The Mirror Clips on the MSA would interfere with a vacuum gripper design. Also, since there are several different MSA optic sizes, there would need to be different vacuum plates for each optic size. It was clear that using the 4 mounting lugs on the sides of the MSAs was the best choice. The mounting lugs for each MSA are in the same location. The design constraints did not leave too many options for the design of the side mounting plates. The plate geometry was completely driven by the tight clearances.
between the MSAs and the LRU frames. Stainless Steel was chosen as the plate material primarily because of its strength and durability over aluminum. Stainless steel also particulates less than aluminum. A latching mechanism was incorporated into the side mounting plates so that a positive capture of the MSA is assured. The side mounting plates consist of three pieces sandwiched together. The two outer pieces are welded together and the middle piece is free to slide in an out. This middle piece is pulled back while engaging the MSA and then pushed forward to lock the MSA in the plate hooks while the End-Effect is handling the MSA.

Because the MSAs vary in width, the distance between the plates needs to be adjustable. There were two conceptual designs considered. The first design concept was to have two parallel rods that the side mounting plates could slide freely on. The side mounting plates would attach to a block that would have a bearing sleeve in them. The rods would be placed through the bearing sleeves and the side plates could simply be pushed or pulled along the rods into the correct position. In order to keep an MSA from moving once it was engaged, clamps would be placed on the rails. The second design considered was a rail and lead screw design. With this design, the side mounting plates could slide in and out on a set of linear rails. A lead screw would be used to drive one of the side plates and the other one would be able to slide freely. A decision matrix was used to choose the best design. This matrix can be found in Table 1.

<table>
<thead>
<tr>
<th>Design Criterion</th>
<th>% Weight of Each Criterion</th>
<th>Criterion Value</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>Rod and Clamp design</td>
</tr>
<tr>
<td>Cost</td>
<td>10%</td>
<td>5</td>
</tr>
<tr>
<td>Safety / Risk to the optic and personnel</td>
<td>40%</td>
<td>2</td>
</tr>
<tr>
<td>Ease to operate and control the motion</td>
<td>30%</td>
<td>2</td>
</tr>
<tr>
<td>Particle generation / need for lubrication</td>
<td>10%</td>
<td>3</td>
</tr>
<tr>
<td>Wear over time</td>
<td>10%</td>
<td>3</td>
</tr>
<tr>
<td><strong>Total</strong> (Sum of % weight x the criterion value)</td>
<td>2.5</td>
<td><strong>3.8</strong></td>
</tr>
</tbody>
</table>

As shown in the decision matrix, the rail and lead screw concept was the better design choice. I chose to use THK stainless steel linear rails and mounts. I was able to work with the vendor to have them use Krytox LVP grease, which is a cleanroom approved lubricant. In choosing the lead screw I needed to find a screw and nut combination that would require little to no lubrication, would last a long time, and would be made out of the acceptable cleanroom materials. This was much harder task than I had expected. I wanted to find something off the shelf because it would most likely be less expensive than designing and fabricating something from scratch. I finally found a Precision Modified-ACME Lead screw and nut combination from McMaster-Carr. The screw is made of 303 stainless steel and the nut is a polyacetal, self lubricating, wear-compensating nut. The material of the nut had not been approved for use in the OAB. I worked with the NIF materials group and received approval for the use of this material for my application. To use the nut, they required that I put a particle catcher around the nut. I also added protective shields near the face of the optic to help mitigate contamination on the mirror surface.
The rail plate of the Rail Sub-Assembly was designed to hold the THK rails and the lead screw as well as interface with the Tilt Sub-Assembly. The width of the plate was chosen based on the widest MSA. The height of the plate was chosen based on the appropriate spacing of the THK rails. As the distance between the rails increases, the load on each rail decreases, but the overall weight of the plate increases. The plate height selected resulted in safety factors well above the required value and an acceptable weight for the Rail Sub-Assembly. Aluminum was chosen as the material for the rail plate to help keep the weight of the End-Effector down. Large cutouts were also placed in the plate to reduce the weight.

There are 4 aluminum blocks that attach to the rail plate to hold the lead screw. These are called the bearing housings and are shown in Figures 6 and 7. The thread of the lead screw was turned down at the location of the four blocks. Peek bushings were placed in each block to lower the friction between the lead screw and the block and to reduce the particle generation. Peek was chosen over other bearing materials because of its low coefficient of friction and durability. E-clips were placed at the two end blocks to prevent the lead screw from translating. There is one other block that holds the lead screw nut. This aluminum block is called the nut housing and is shown in Figure 6. It is important that a majority of the MSA weight is transferred to the THK rails and not onto the lead screw. To transfer the load correctly, the nut needs to be perfectly aligned with the lead screw both when the End-Effector is loaded and unloaded. To minimize the load transferred to the lead screw, Belleville washers were placed between the nut housing and the bar it is mounted to.

The last pieces of the Rail Sub-Assembly are the two hinges shown in Figure 7. These are used to interface with the Tilt Sub-Assembly and assist with the ±20° yaw rotation. This will be discussed further in the next section.

![Figure 7 Additional components of the Rail Sub-Assembly](image-url)
**Tilt Sub-Assembly Design**

The main function of the Tilt Sub-Assembly is to safely rotate the MSA in pitch and yaw. Like the Rail Sub-Assembly, the main constraint of the Tilt Sub-Assembly design was the size. Since the total length of the End-Effector is limited to 495 mm, the total length of the Tilt Sub-Assembly had to be minimal. The other constraint that played a large role in this part of the design was the requirement to be able to safely control the angular motion. Lastly, the weight was a consideration because of the maximum moment allowed on the NOID Robo-hand.

There were several conceptual designs considered for the Tilt Sub-Assembly, which will be discussed later in this section. The final design is shown in Figures 8 and 9. The main components of the Tilt Sub-Assembly are the two parallel aluminum plates, the rotating pins and hinges, the pitch adjustment system, and the yaw locking system.

![Diagram of Tilt Sub-Assembly Components]

**Figure 8** Components of the Tilt Sub-Assembly

![Diagram of Pitch Adjustment System]

**Figure 9** Pitch adjustment system of the Tilt Sub-Assembly
When an MSA is being installed in an LRU, the MSA must be parallel to the mating surface on the LRU. In order for this to happen, the yaw orientation of the MSA relative to the NOID arm must continually be changing as the NOID arm is extended. For this reason, the yaw rotation of the End-Effector must be easy to control and adjust. The pitch rotation, on the other hand, only needs to be adjusted once or twice during the installation of an MSA.

To keep the design costs down, the first idea for this portion of the design was to see if there was anything commercially available that would meet the requirements. Unfortunately, I did not find anything that would meet the space, material, and load requirements of this system. The next idea was to use a combination of a commercially available joint with a custom design. There were two joint options: I could use a ball and socket joint, which would allow rotation in all directions or I could use a universal joint that would only allow pitch and yaw rotation. In both cases I would need to design mechanisms that would safely control the rotation. A sketch of this design idea is shown in Figure 10. The other idea was to design a mechanism that would control the pitch rotation while allowing the End-Effector to freely rotate in yaw about a vertical pin. I came up with several mechanisms that would adequately adjust and control the pitch motion. Two of the mechanisms considered are shown in Figure 10. A decision matrix was used to choose the best conceptual design, as shown in Table 2.

![Figure 10 Tilt Sub-Assembly design idea sketches](image-url)
Table 2 Decision matrix for the mechanism to adjust the pitch and yaw
(Each criterion was given a value between 1 and 5, with 1 being the lowest and 5 being the highest. The design with the highest total was deemed the best overall design.)

<table>
<thead>
<tr>
<th>Design Criterion</th>
<th>% Weight of Each Criterion</th>
<th>Criterion Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Total design and manufacturing cost</td>
<td>15%</td>
<td>Design using a commercially available joint Pitch Mechanism Design</td>
</tr>
<tr>
<td>Safety / Good motion control</td>
<td>35%</td>
<td>4</td>
</tr>
<tr>
<td>Particle generation / need for lubrication</td>
<td>15%</td>
<td>3</td>
</tr>
<tr>
<td>Wear over time</td>
<td>10%</td>
<td>3</td>
</tr>
<tr>
<td>Light weight desired</td>
<td>10%</td>
<td>4</td>
</tr>
<tr>
<td>Short length desired</td>
<td>15%</td>
<td>4</td>
</tr>
<tr>
<td><strong>Total</strong> (Sum of % weight x the criterion value)</td>
<td></td>
<td><strong>3.4</strong></td>
</tr>
</tbody>
</table>

From the decision matrix it was clear that the joint design was best, however, after looking at the commercially available joints I decided that it would be better to design a custom universal joint that would better fit my application. The pitch and yaw plate assembly previously shown in Figure 8 is the universal joint that I designed. This design is significantly more compact than any of the designs I could come up with using a commercially available joint. The two vertically oriented pins interface with the hinges and flanged peak bearings on the Rail Sub-Assembly. These pins allow for the yaw rotation. I decided to leave the yaw rotation free so that the MSA could be easily adjusted by hand while the MSA is being installed in the LRU. I added a locking system to the top pin so that the yaw rotation could be held if desired. The locking system was added later and is therefore not shown in the model. It can be seen in Figure 14 on page 18. The top vertical pin was made longer and was threaded on the top end. The two threaded rings of the locking system were designed to act like two locking nuts. The friction of the bottom ring against the top hinge is what prevents the Rail Sub-Assembly from rotating in Yaw when the locking system is engaged.

The two horizontal pins allow for the pitch rotation of the End-Effector. The last part of the Tilt Sub-Assembly to be designed was the mechanism to adjust and control the pitch rotation. There were two options that were considered. The first idea was to use a threaded rod through one plate that would contact the second plate near the bottom. With this design gravity would push the second plate against the threaded rod. As the rod is threaded in or out, the plate would tilt up or down. The second idea also used a threaded rod through one plate, but this time the rod would be attached to the second plate. For this second design to work the threaded connection in the first plate would have to be able to rotate and the connection with the second plate would have to mimic a ball and socket joint. The second design is safer than the first because it would positively connect the threaded rod to both plates, but it is significantly more complicated. I ended up selecting the first design, because the added safety of the second design was not worth the extra design complications. The final pitch adjustment system is shown in Figure 9. The hand wheel is attached to a fine pitch threaded rod which is threaded through a stainless steel keeninsert in the aluminum plate. Since it is common for similar materials to gall together in a cleanroom, brass was selected for the threaded rod material. Cleanroom approved lubrication
was also placed on the keensert to lower the friction and allow for easier turning of the handle. A lot of thought was put into the end of the threaded rod that would slide up and down the second plate as the threaded rod is turned. Particle generation, contact stresses, and friction were all factors in this part of the design. A Tungsten Carbide ball from McMaster-Carr was inserted into the end of the threaded rod. A modified Tungsten Carbide bushing from Reid Tool was chosen as the contact surface for the ball. The bushing was cut in half and glued to the second plate, providing a curved surface for the ball to slide against. A curve surface was selected over a flat surface to reduce the contact stresses. Tungsten Carbide was chosen because of its strength, low friction, and wear resistant properties. Other Tungsten Carbide designs were considered, but they would have required new parts to be designed and manufactured. The ball and modified Tungsten Carbide bushing design was chosen because the parts were commercially available and inexpensive.

Twist Sub-Assembly Design
The main function of the Twist Sub-Assembly is to provide a controlled twist rotation of the MSA. The other function of this sub-assembly is to attach the whole end-effector to the NOID Robo-hand. The main design considerations of the Twist Sub-Assembly design were the size, the load capacity, and safety. Since the Twist Sub-Assembly is furthest from the MSA, there is a large moment load it must withstand.

There were several conceptual designs considered for the Twist Sub-Assembly, which will be discussed later in this section. The final design is shown in Figure 11. The main components of the Twist Sub-Assembly are the aluminum shaft housing, the shaft, two roller bearings, a clamp collar, and the twist mechanism.

![Twist Sub-Assembly Components](image)

Figure 11 Twist Sub-Assembly Components
There were four conceptual designs considered for the Twist Sub-Assembly. Sketches of the four concepts are shown in Figure 12. The first concept is the most simple. It simply uses bolts, washers, and a plate with arced slots. The bolts would always be tight until a twist adjustment is required, at which time the bolts would be slightly loosened and the End-Effector (with an MSA) would be rotated by hand. This design concept is simple, but it has some very apparent safety concerns. If the MSA was not centered on the Rail Sub-Assembly it is possible that the End-Effector would twist on its own if the bolts were not tight enough. Also, when the bolts are loose it is risky to have personnel control the twist by hand. The second design concept considered is also very simple. It uses a large threaded shaft that would turn inside a shaft housing as shown in Figure 12. There would be a locking nut to hold the shaft from rotating when twist rotation was not desired. When twist rotation is required, the nut would be loosened and then the End-Effector (with an MSA) would be rotated to the desired position and the nut would be tightened again. This design is simple, but the process could be cumbersome if several iterations were required to get the exact twist orientation required. The design also has safety issues when the locking nut is loosened. The third concept uses a worm gear assembly to rotate a shaft that is connected to the rest of the End-Effector, also shown in Figure 12. This design provides a safer and more controlled twist adjustment. The challenges with this design were to find gears that are cleanroom compatible, provide the fine adjustment required, and have minimal backlash. The last design concept consists of a shaft, a shaft housing, and a mechanism that will cause the shaft to rotate. At this point in the design, there were a few design ideas for the mechanism that would control the twist rotation, but nothing had been finalized. Once again the pros and cons of each concept were considered and a design was chosen. The pros and cons are summarized in the decision matrix found in Table 3.

Figure 12 Twist Sub-Assembly design idea sketches
Table 3 Decision matrix for the twist design concept
(Each criterion was given a value between 1 and 5, with 1 being the lowest and 5 being the highest. The design with the highest total was deemed the best overall design.)

<table>
<thead>
<tr>
<th>Design Criterion</th>
<th>% Weight of Each Criterion</th>
<th>Criterion Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Total design and manufacturing cost</td>
<td>10%</td>
<td>5</td>
</tr>
<tr>
<td>Safety / Risk to the optic and personnel</td>
<td>30%</td>
<td>1</td>
</tr>
<tr>
<td>Smooth and easy operation under a large moment load</td>
<td>20%</td>
<td>2</td>
</tr>
<tr>
<td>Fine adjustment</td>
<td>15%</td>
<td>1</td>
</tr>
<tr>
<td>Cleanliness: Particle generation / need for lubrication</td>
<td>5%</td>
<td>4</td>
</tr>
<tr>
<td>Wear over time/ backlash</td>
<td>10%</td>
<td>3</td>
</tr>
<tr>
<td>Short length desired</td>
<td>10%</td>
<td>5</td>
</tr>
<tr>
<td><strong>Total</strong> (Sum of % weight x the criterion value)</td>
<td></td>
<td><strong>2.35</strong></td>
</tr>
</tbody>
</table>

As shown in Table 3, the worm gear and twist mechanism designs were far superior. Even though the twist mechanism design required more design effort, it was the superior design concept because of the need for fine adjustment and minimal backlash. After that concept was chosen, the twist sub-assembly had to be designed. First, I focused on the shaft and the shaft housing. The shaft design was mostly driven by the moment load it would carry and the size of the bearings that would support the shaft. The detailed drawing of the shaft can be found in Appendix A. The face of the shaft is 4.5 inches in diameter and interfaces directly with the Tilt Sub-Assembly. This way, when the shaft is rotated the tilt and rail sub-assemblies will also rotate. The shaft diameter steps down twice, once to fit the inner diameter of the larger bearing and again to fit the inner diameter of the smaller bearing. A radius was added at each step down to reduce the stress concentrations in the shaft. The end of the shaft was threaded so that a threaded clamp collar could be placed at the end of the shaft to hold the bearings in place. The two bearings in the Twist Sub-Assembly are tapered roller bearings from Timken. These were lubricated with cleanroom approved lubrication. I chose tapered roller bearings because they can take both thrust and radial loads. The threaded clamp collar was from King Bearing. The bearings and the clamp collar are concealed in a shaft housing. The main functions of the shaft housing are to support the shaft bearings and to connect the whole End-Effector to the NOID Robo-hand. The detailed drawing of the shaft housing can be found in Appendix A. The length of the shaft housing was chosen to give the whole End-Effector the optimal length for installing the MSAs into the LRUs. The internal cut outs were sized to properly hold the bearings and to allow for easy assembly of the Twist Sub-Assembly. The last component needed for this Sub-Assembly was the mechanism that would be used to turn the shaft.

After a lot of brainstorming and several sketches I came up with the twist mechanism shown in Figure 13. The twist mechanism consists of a threaded brass rod that is threaded through a slightly lubricated keensert in the shaft housing. As the threaded rod is turned, it pushes/pulls a fork which then pushes/pulls the force ring, causing the shaft to rotate.
The threaded rod is made from brass and has a ½-20 fine pitch thread. Brass was chosen because it is unlikely to gall with the stainless steel keenert. The fine pitch thread was chosen to allow for fine adjustments of the twist rotation. The threaded rod is turned down where it connects to the aluminum fork as shown in Figure 13. There is a flanged peak bearing sleeve that goes through the fork and allows the rod to turn in the fork. Peak was chosen for the bearing sleeve because of its low friction and wear resistant properties. The rod is held in place with two E-clips. A Delrin washer was placed between the E-clip and the fork on the non-flanged side of the peak bearing. On the other end of the force ring there is a stainless steel dowel pin that is pressed into the fork and goes through the vertical slot in the force ring. This dowel pin has a peak bearing sleeve around it that allows for a low friction interaction between the dowel pin and the force ring as the pin moves up and down in the slot. The peak sleeve was selected so that inner diameter just fit over the dowel pin. The slot in the force ring was sized so that there was very little slop between it and the peak sleeve. This was carefully done to minimize the backlash in the twist mechanism assembly. There are also two Delrin washers on the dowel pin axle that were placed on each side of the force ring. These washers were placed there so that the aluminum force ring did not side against the aluminum fork and create particles. The last piece of the twist mechanism is the stainless steel key that locks the force ring in position against the shaft.
The Design as a Whole

The last components to be selected were the hand wheels. The Rail Sub-Assembly, Tilt Sub-Assembly, and Twist Sub-Assembly each need one hand wheel. The hand wheel sizes were selected based on the clearance between the other components and on the estimated amount of force required to turn the wheel when the End-Effector is holding an MSA. All hand wheels were purchased from Carr Lane and are made of stainless steel. The hand wheel for the Rail Sub-Assembly is attached in such a way that it can easily be removed and moved to either side of the lead screw as required.

With the hand wheels in place and all three Sub-Assemblies mated together, the MSA NOID End-Effector design is complete and all design requirements have been met. The completed End-Effector is shown installed on the NOID in Figure 14.

![Figure 14 MSA NOID End-Effector installed on the NOID arm](image)

As stated earlier, I modeled the MSA NOID End-Effector design in Pro/DESKTOP. I used the End-Effector assembly model to perform an interface study to make sure that in all cases the MSA NOID End-Effector could install the MSAs without interfering with the LRU frames. The clearance is tight in some cases, but the study showed that the appropriate MSAs can be installed in each LRU type. Figure 15 shows some of the tight clearances that were found in the study. As can be seen in the figure, the Rail Sub-Assembly hand wheel would have to be moved to a specific side of the lead screw in order for the MSA to be installed into an LM4 LRU.
Figure 15 MSA NOID End-Effector interface with an LM4 (upper) and an LM7 (lower)

**Design Calculations**

Before any equipment designed at Lawrence Livermore National Laboratory can be used, a safety note must be written and approved. The safety note for the MSA NOID End-Effector addresses the structural integrity of the End-Effector. The safety note calculations can be found in Appendix B. The End-Effector was designed per guidelines of the Lawrence Livermore National Laboratory Mechanical Engineering Design Safety Standards. In most cases the Design
Safety Standards require a minimum safety factor of 3 on yield or 4 on ultimate for static loads and a minimum safety factor of 1 on yield or 1.25 on ultimate for seismic loads.

The largest MSA weight of 270 lbs was used for the calculations. When the MSA is cantilevered off of the Side Mounting Plates, an over turning moment is created throughout the assembly. Overall, the machined components were considered very robust and the critical load path was assumed to be through the fasteners connecting the components and through the shaft of the Twist Sub-Assembly. It was also conservatively assumed that the MSA was rotated 20° in yaw and was 4 inches off center along the THK rails on the Rail Sub-Assembly. This placed an additional shear load on most of the End-Effector components.

Both static and seismic calculations were performed on the End-Effector. For the seismic calculations it was assumed that the LM4-8 NOID End-Effector will be subject to 1.3g vertical and 1.3g horizontal loads during a seismic event. It is unrealistic to have 100% of the seismic load applied to all three directions (X, Y, and Z) at the same time. For this reason, NIF seismic guidance was applied to the calculations. That is, vertical seismic load (Y) =100%, horizontal seismic loads (X and Z) = 40%. Due to the free-swinging nature of the NOID arm around the center column, horizontal accelerations in “Z” were considered nil.

As can be seen in Table B1 of Appendix B, most of the MSA NOID End-Effector components met the Design Safety Standards. The few that did not meet the Design Safety Standards were accepted based on the improbability of occurrence or acceptable consequences of the component yielding. The worst case static moment load on the Robo-hand was calculated to be 5319 in-lb which is just under the manufactures rated load of 6000 in-lb.

**Procure, Assemble, and Test**

The drawings found in Appendix A were sent to several machine shops for bidding. Lukas Machine, in Seattle Washington had the lowest bid at $4409.00 and was awarded the job. When the cost of the machined components is totaled with the cost of the “off the shelf” items, the total manufacturing cost of the MSA NOID End-Effector was just under $6,000, which met the low cost requirement of the design.

Once all the components were received, a dirty fit up was performed to make sure that all the pieces fit together as designed. At the time of the dirty fit up, a new yaw locking system was designed and fabricated. The parts were then disassembled and precision cleaned. The final assembly was performed in the OAB and the assembly was attached to the NOID as shown previously in Figure 14.

The final step before the MSA NOID End-Effector could be used for production was a load and function test. The load test was performed with 405 lbs hung from the End-Effector at the location of the MSA’s center of gravity, as shown in Figure 16. The 405 lbs represented 150% of the weight of the heaviest MSA. The test was performed with the End-Effector at the extreme twist and yaw angles and with the MSA weight off center by 4 inches on the Rail Sub-Assembly. The load was held for 10 minutes and then the End-Effector was inspected for yielding. During the inspection it was discovered that the E-clips that were used in the Twist Sub-Assembly were not strong enough to carry the load at the extreme angles. The E-clips were replaced with a set
screw design and the assembly was reload tested. There were no signs of yielding during the second load test.

The function test of the End-Effector followed the load test. The function test was first performed using aluminum blanks for optics and installing the blank MSAs into an LM4 LRU frame. The second function test was performed using practice optics and installing the practice MSAs into an LM5 LRU frame. A picture of this function test is shown in Figure 17. Due to hardware restraints at the time a fit test was not performed with an LM6, LM7, or LM8 LRU. Since the designs are very similar, there is not expected to be any problems installing the MSAs into these LRUs.

Figure 16 Load test of the MSA NOID End-Effector

Figure 17 Fit and function test of the MSA NOID End-Effector with an LM5 Practice optic
Conclusion

The final testing of the MSA NOID End-Effector was completed in June, 2004. Soon after the testing was complete, the LM4-8 LRU production was put on hold for a couple of years while other parts of the NIF laser were built. It was only in October, 2006 that the End-Effector was used to build its first production LM5 LRU. It is now December, 2006 and the End-Effector is currently being used to build a production LM4 LRU. So far the End-Effector has worked well and has been successful in installing 8 MSAs. LM6, LM7, and LM8 LRUs are scheduled to be assembled in January, 2007. I have no doubt that the MSA NOID End-Effector will effectively install the MSAs into these LRUs.

Appendices

Appendix A: MSA NOID End-Effector component drawings.

Appendix B: MSA NOID End-Effector Safety Note Calculations
Appendix A-Reference Drawings
26 attached pages
NOTES:
1. GROSS CLEAN
2. REMOVE ALL BURRS AND BREAK ALL SHARP EDGES
3. 1.6 [63 MICROINCHES] FINISH
4. TOLERANCES ARE ±0.5mm [0.020in] UNLESS NOTED OTHERWISE
3X M4 Ø16 ±0.63 ±0.04in
3X Ø28.56 ±1.15in
2X Ø30 ±0.017in
2X Ø32 ±0.017in

2X 45° ±1° X 3
0.125in

Ø10.0 THRU
Ø5.79in

2X M9 Ø19 ±0.748 ±0.04in

1.0" STK
12.70
0.500in
18.57 ±0.12
0.731 ±0.005in
18.58 ±0.12
0.731 ±0.005in

NOTES:
1. GROSS CLEAN
2. REMOVE ALL BURRS AND BREAK ALL SHARP EDGES
3. 1.6 (63 MICROINCHES) FINISH
4. TOLERANCES ARE ±0.3mm (±0.012in)
   UNLESS NOTED OTHERWISE
NOTES:
1. REMOVE ALL BURRS AND
2. BREAK ALL SHARP EDGES
3. 1.6 [63 MICRONCHES] FINISH
4. TOLERANCES ARE ±0.5mm [0.020in]
UNLESS NOTED OTHERWISE

2X Ø55 ±0.1 THRU
Ø0.217 ±0.004 in
10.0 ±1 \( \times \) 19.1 ±1
0.36 ±0.004 \( \times \) 0.75 ±0.04 in
Ø0.200 ±0.005 in

2X 45° 12° x 3 ±0.12 ±0.04 in
NOTES:
1. GROSS CLEAN
2. REMOVE ALL BURRS AND
   BREAK ALL SHARP EDGES
3. 1.6 [63 MICRONCHES] FINISH
4. TOLERANCES ARE ±0.5mm [0.020in]
   UNLESS NOTED OTHERWISE
NOTES:
1. CROSS CLEAN
2. REMOVE ALL BURRS AND
   BREAK ALL SHARP EDGES
3. 1.6 [63 MICRONCHES] FINISH
4. TOLERANCES ARE ±0.05 mm [0.020 in]
   UNLESS NOTED OTHERWISE
NOTES:
1. GROSS CLEAN
2. REMOVE ALL BURRS AND BREAK ALL SHARP EDGES
3. 1.6 [63 MICRONCHES] FINISH
4. TOLERANCES ARE ±0.3mm [0.012in]
   UNLESS NOTED OTHERWISE

- 4X Ø12.685 ±0.005 ±19.05
- 4X Ø0.4984 ±0.0002 ±0.75
- 4X 85.35 MAX
- 4X 45°±2° X 2 [0.079in]
NOTES:
1. GROSS CLEAN
2. REMOVE ALL BURRS AND
   BREAK ALL SHARP EDGES
3. 1.6 [35 MICROINCHES] FINISH
4. TOLERANCES ARE ±0.5 mm [0.02 in]
   UNLESS NOTED OTHERWISE

\[ \phi 5.540 \pm 0.008 \text{ in} \quad \phi 0.2181 \pm 0.0003 \text{ in} \]
\[ 2.00 \pm 0.05 \quad 0.079 \pm 0.002 \text{ in} \]

- 3/8'-24 FULLY THREADED ROD
1. GROSS CLEAN
2. REMOVE ALL BURRS AND BREAK ALL SHARP EDGES
3. 15 [63 MICRONOSES] FINISH
4. KEY SLOT MUST BE PLACED IN POSITION SHOWN W.R.T THE THREADED HOLES
5. TOLERANCES ARE ±0.03mm [0.012in] UNLESS NOTED OTHERWISE

STANDARD THREAD RELIEF

6X M8 X 1.25-6H THRU CH @ 88.9 [3.50in]

SHAFT
1/2"-13 THREAD
+0.100
-0.000
+0.004
-0.000
∅0.375 - 0.000

4X R0.2 MAX
R0.0008

25.4 ± 1.0
1.00 ± 0.04
+0.23
12.98 ± 0.00
-0.009
0.511 ± 0.0006

∅7.696 ± 0.076
+0.001
∅0.303 ± 0.003

45° ± 2° X 1 [0.04°]

∅1 ± 0.05
0.20 ± 0.02

R1.0 ± 0.05
R0.04 ± 0.02

NOTES:
1. CROSS CLEAN
2. REMOVE ALL BURRS AND
3. 1.6 [63 MICRONCHES] FINISH

BREAK ALL SHARP EDGES
Appendix B-Supporting Calculations
55 attached pages

Weight Definitions:
$W_1 = 276$ lb (Weight of largest MSA + Side Mounting Plates)
$W_2 = 6$ lb (Weight of 2 LM Mounts)
$W_3 = 26.5$ lb ($W_2 +$ Weight of THK Rails, LM Blocks, Threaded Rod, and Rail Plate)
$W_4 = 3$ lb (Weight of Tilt Plate and Hinges)
$W_5 = 6.7$ lb ($W_4 +$ Weight of Pitch Plate and Pitch Crank)
$W_6 = 17.9$ lb (Weight of Shaft + Adapter Block)
<table>
<thead>
<tr>
<th>Component Description</th>
<th>Failure Mode</th>
<th>S.F. Static</th>
<th>S.F. Dynamic</th>
</tr>
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<td><strong>Rail Sub-Assembly</strong></td>
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<td>Side Mounting Plates</td>
<td>Bending stress</td>
<td>133.3</td>
<td>48.1</td>
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<td>Screws securing Side Mounting Plates to the LM Mounts (M5)</td>
<td>Von Mises Bolt Stress</td>
<td>8.3</td>
<td>2.9</td>
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<tr>
<td>Screws securing the LM Blocks to the LM Mounts (M4)</td>
<td>Von Mises Bolt Stress</td>
<td>7.9</td>
<td>3.1</td>
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<td>THK Rails</td>
<td>Manufacturer’s rated load</td>
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<td>7.3</td>
<td>2.8</td>
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<tr>
<td></td>
<td>Al thread stripping</td>
<td>18.4</td>
<td>N/A</td>
</tr>
<tr>
<td>Screws securing the Hinges to the Back Plate (M8)</td>
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<td>5.8</td>
<td>2.1</td>
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<td></td>
<td>Al thread stripping</td>
<td>27.7</td>
<td>10.0</td>
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<td><strong>Tilt Sub-Assembly</strong></td>
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<td>Tilt Plate Pins</td>
<td>Von Mises Stress</td>
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<td>1.6</td>
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<td>Screws securing the Hinges to the Pitch Plate (M8)</td>
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<td>1.6</td>
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<td>Threaded Rod through the Pitch Plate</td>
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<td>13.5</td>
<td>5.0</td>
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<tr>
<td></td>
<td>Max force to turn handle</td>
<td>13.1 lbf</td>
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</tr>
<tr>
<td>Tungsten Carbide ball</td>
<td>Contact stress</td>
<td>1.1</td>
<td>N/A</td>
</tr>
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<td><strong>Twist Sub-Assembly</strong></td>
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<td></td>
<td></td>
</tr>
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<td>Screws securing the Pitch Plate to the Shaft (M8)</td>
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<td>4.4</td>
<td>1.6</td>
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<td>Shaft Key</td>
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<td>1.8</td>
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<td>Force Ring</td>
<td>Shear at the slot</td>
<td>2.9</td>
<td>1.1</td>
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<td>E-Clip</td>
<td>Shear</td>
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<td>Screws securing the Adapter Block to the Robohand (M8)</td>
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<td></td>
<td>Al thread stripping</td>
<td>9.4</td>
<td>3.6</td>
</tr>
</tbody>
</table>

1. Ultimate Strength used, in most cases the Ultimate Strength is all the manufacturer provided
2. Not a critical part and yielding would be acceptable
3. Calculation was not needed, see notes within the calculations
4. Values are slightly lower than the Safety Factors recommended in the MEDSS, but the stress on these components are primarily a result of a twist moment caused by the unlikely event of the MSA being 3” off center on the rails
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<thead>
<tr>
<th>Part</th>
<th>Sub-Assembly</th>
<th>Quantity</th>
<th>Manufacturer</th>
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<td>McMasters</td>
<td>Brass, 1/2-20 Thread</td>
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<td>Keensert</td>
<td>Twist</td>
<td>1</td>
<td>LLNL Shop</td>
<td>SST, 1/2-20 Inner Thread</td>
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<td>Delrin Washer</td>
<td>Twist/Rail</td>
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<td>McMasters</td>
<td>3/8&quot; ID, 0.625 OD, 0.031&quot; THK</td>
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<td>E-Clips</td>
<td>Twist/Rail</td>
<td>4</td>
<td>McMasters</td>
<td>3/8&quot; Stainless Steel</td>
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<td>THK</td>
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<td>Set Screws</td>
<td>Twist</td>
<td>8</td>
<td>Danforth</td>
<td>10-32 x 3/16&quot;, 1/2 dogend, Heat-treated alloy steel</td>
</tr>
</tbody>
</table>
**Safety Calculations.**

Heaviest MSA = 270 lb (FABS)

Load Path:

<table>
<thead>
<tr>
<th>Part</th>
<th>QTY</th>
<th>Weight (lb)</th>
<th>Bg = &amp; Calc</th>
<th>NOTES</th>
</tr>
</thead>
<tbody>
<tr>
<td>MSA</td>
<td>1</td>
<td>270</td>
<td>N/A</td>
<td>MESNO2-116</td>
</tr>
<tr>
<td>Mounting plate</td>
<td>2</td>
<td>3</td>
<td>2</td>
<td>SST 4130</td>
</tr>
<tr>
<td>LM mount</td>
<td>2</td>
<td>3</td>
<td>7</td>
<td>AL 6061</td>
</tr>
<tr>
<td>LM block</td>
<td>4</td>
<td>0.5</td>
<td>7</td>
<td>THK</td>
</tr>
<tr>
<td>Rail</td>
<td>2</td>
<td>2</td>
<td>10</td>
<td>THK</td>
</tr>
<tr>
<td>Rail block</td>
<td>1</td>
<td>0.5</td>
<td>12</td>
<td>AL</td>
</tr>
<tr>
<td>Threaded rod</td>
<td>1</td>
<td>0.5</td>
<td>12</td>
<td>SST</td>
</tr>
<tr>
<td>Rail plate</td>
<td>1</td>
<td>14</td>
<td>13</td>
<td>AL 6061-76</td>
</tr>
<tr>
<td>HIL-Plate</td>
<td></td>
<td>2.8</td>
<td>18</td>
<td>AL 6061-76</td>
</tr>
<tr>
<td>Pitch plate</td>
<td>1</td>
<td>2.7</td>
<td>21</td>
<td>AL 6061-76</td>
</tr>
<tr>
<td>Pitch crank</td>
<td>1</td>
<td>1.0</td>
<td>26</td>
<td>SST</td>
</tr>
<tr>
<td>Shaft</td>
<td>1</td>
<td>1.9</td>
<td>29</td>
<td>SST 304</td>
</tr>
<tr>
<td>Force ring/bak</td>
<td>1</td>
<td>1.2</td>
<td>41</td>
<td>AL</td>
</tr>
<tr>
<td>Adapter block</td>
<td></td>
<td>1.4</td>
<td>49</td>
<td>AL</td>
</tr>
<tr>
<td>Rods/stand</td>
<td>1</td>
<td>N/A</td>
<td>53</td>
<td>Max moment allowed = 6000 m/lb</td>
</tr>
</tbody>
</table>

**NOTE:** The critical path is through the bolts. To be conservative, the 3rd value for A2-56 bolts will be used, even though most will be A2-70.
Bending stress in the side mounting plates

\[ W = 270.3 \text{ kN} \]
\[ W = 2 \text{ in} \]
\[ \frac{W}{2} = 1.25 \text{ in} \]
\[ h = 13 \text{ in} \]
\[ b = 12 \text{ in} \]

\[ I = \frac{bh^3}{12} = \frac{12 \times 13^3}{12} = 22.88 \text{ in}^4 \]

\[ C = \frac{d}{2} = 6.5 \text{ in} \]

\[ M_b = \frac{W}{2} (6.34) = 855.9 \text{ in}-\text{lb} \]

\[ \sigma_b = \frac{Mc}{I} = \frac{243.1}{22.88} = 10 \text{ ksi} \]

\[ S_y \geq 30 \text{ ksi} \]

\[ F_3 = \frac{S_y}{\sigma_b} = \frac{30,000}{243.1} = 123.4 > 3 \text{ ... OK} \]

Seismic:

Apply 100-40-40 rule. Due to the fact that the N-D arm is free swinging in Z, assume Z accelerations are negligible.
SDy = W : C_p : I_p : Acc : N_p (40%)
SDz = W : C_p : I_p : Acc : N_p (100%)
Acc = 1.3g (seismic acceleration)
C_p = 1 (base factor)
I_p = 1 (importance factor)
N_p = 1.2 (correction factor PC1/PC2)

SDx = 270 lb x 1 x 1 x 1.3 x 1.2 x 1.4 = 168.5 lb
SDy = 270 lb x 1 x 1 x 1.3 x 1.2 x 1.1 = 421.2 lb

A = b x h = 1.625 in² (plate area)

Stress from SDx:
\[ \sigma_{SDx} = \frac{SD_x}{2A} = 51.8 \text{ psi} \]

Stress from SDy:
\[ \sigma_{SDy} = \frac{SD_y}{2} \times 6345 = 1.33 \times 10^3 \text{ in lb} \]
\[ \sigma_y = \frac{\sigma_{SDy} 	imes C}{I} = \frac{1.33 \times 10^3 \times 6.5}{22.885} = 379.2 \text{ psi} \]

Combine static & seismic stresses:
\[ \sigma_{total} = \sigma_y + \sigma_{SDx\, 1\, s} + \sigma_0 = 674.2 \text{ psi} \]

FS = \[ \frac{\sigma_{SDy}}{\sigma_{total}} = \frac{30,000 \text{ psi}}{674 \text{ psi}} = \frac{44.5}{1.0 \therefore \text{OK}} \]
Look at the tangs bending + shearing due to SDx

\[ F = \frac{SDx}{4} \text{ for 4 lug total} \]

\[ SDx = 168.5 \text{ lb (from pgs 3)} \]

\[ F = \frac{168.5}{4} = 42.13 \text{ lb} \]

Worse-case, \( F \) would be at the top of the tang.

\[ C = \frac{5 \text{ mm}}{2} = 2.5 \text{ mm} = 0.098'' \]

\[ A = 5 \text{ mm} \times 3.175 \text{ mm} = 15.875 \text{ mm}^2 = 0.026 \text{ in}^2 \]

\[ I = \frac{bh^3}{12} = \frac{3.175(5)^3}{12} = 33.07 \text{ mm}^4 = 7.95 \times 10^{-5} \text{ in}^4 \]

\[ M = F \times \frac{9 \text{ mm}}{25.4 \text{ mm}} = 42.13 \times \frac{9}{25.4} = 14.93 \text{ in lb} \]

\[ \sigma = \frac{Mc}{I} = \frac{14.93 \text{ in lb} \times 0.098\text{in}}{7.95 \times 10^{-5} \text{ in}^4} = 18.41 \text{ ksi} \]

\[ \tau = \frac{3}{2} \frac{F}{A} = \frac{3}{2} \frac{4213 \text{ lb}}{0.026 \text{ in}^2} = 2.53 \text{ ksi} \]

Shear of a rectangle

\[ G_{\text{max}} = \sqrt{\sigma^2 + 3 \tau^2} = \sqrt{18.41^2 + (2.53)^2} = 18.92 \text{ ksi} \]

\[ S_y = 30 \text{ ksi} \]

\[ F.S. = \frac{30.0}{18.92} = 1.59 \geq 1 \text{ -- ok} \]
Side Platen (shear in screws)

\[ W = MSA \text{ weight} + 2 \times \text{side plate weight} \]

\[ W = 270 + 6 = 276 \text{lb} \]

A moment is created which places a shear stress on the bolts.

\[ F = \frac{W \times \frac{8}{12.5}}{12.5} = 88.3 \text{ lb} \]

For both \( F + V \), assume that only two screws carry the load due to misalignment.

\[ V = \frac{W}{2} = 67.5 \text{ lb} \]

Total force on screw = \[ \sqrt{88.3^2 + 67.5^2} = 111.2 \text{ lb} \]

\[ A_s = 14.2 \text{mm}^2 = 0.0022 \text{ in}^2 \]

\[ \sigma = \frac{111.2 \text{ lb}}{0.022 \text{ in}^2} = 5.05 \text{ ksi} \]

Because the bolts in shear: \[ S_{kt} = \frac{72.5 \text{ ksi}}{5.05 \text{ ksi}} = 14.3 \text{ ksi} \]

Note: 72.5 ksi, but came from Fabrey catalog for A2-50 SSt bolts.

\[ FS = \frac{41.9 \text{ ksi}}{5.05 \text{ ksi}} = 8.3 \text{ on ultimate} \]

Bolt shear under seismic load:
Acc = 11.3 g

For 3 assembly NOID amplification using 100-40-40 Rule:

\[ C_p = 1 \quad N_p = 1.2 \quad I_p = 1 \]

\[ \left( \frac{D}{S} \right)_f = \text{same for all future calculations} \]

\[ SD_x = W \cdot C_p \cdot I_p \cdot Acc \cdot N_p \cdot (40\%) = 2.76(1)(1)(1.3)(1.2)(0.4) = 17.2 \text{ lb} \]

\[ SD_y = W \cdot C_p \cdot I_p \cdot Acc \cdot N_p \cdot (100\%) = 2.76(1)(1)(1.3)(1.2)(1) = 43.056 \text{ lb} \]

Note: Do not neglect SD_y because the NOID gantry is free in that direction. The friction not of the assembly prevents rotation of the rail subassembly.

Load from SD_y:

\[ F_y = \frac{SD_y \times 8}{12.5} = 137.8 \text{ lb} \]

\[ V = \frac{SD_y}{2} = 107.6 \text{ lb} \]

Load from SD_x:

Assume load is taken by only 2 screws

\[ \frac{SD_x}{2} = 43.1 \text{ lb} \]

Worst case screw:

\[ \sqrt{\left( 137.8 + 88.3 + 43.1 \right)^2 + \left( 107.6 + 67.5 \right)^2} = 321.1 \text{ lb} \]

\[ \sigma = \frac{321.1 \text{ lb}}{0.022 \text{ in}^2} = 141.60 \text{ ksi} \]

\[ FS = \frac{41.9 \text{ ksi}}{141.6 \text{ ksi}} = 2.9 \]

> 1.25; ok
Torque for Side Plate Bolts:

\[ T = K \cdot D_n \cdot F_p \]

\( K = 0.3 \) (High friction in cleanroom)

\( D_n = 0.171 \text{ in} \)

\( F_p = 65\% \left( 72.5 \text{ ksi} \times (0.022 \text{ in}^2) \right) = 1037 \text{ lb} \)

(there is no axial force in this case)

\[ T = 0.3 \cdot (0.171) \cdot (1037) = 53.2 \text{ lb in} \]

\( \text{LM Mount + LM Block} \)

- LM Block 4x (THK product: SHS 15V)
- LM Mount 2x (Aluminum)

Each block has 4 M9 screws threaded into at 6 mm deep.

\[ W_1 = 276 \text{ lb (From pg 5)} \]

\[ W_2 = 2 \times \text{LM Mount} = 6 \text{ lb} \]

Moment created: \[ M = 276 \left( 8.0 \right) + 6 \left( 6.5 \right) = 2211.9 \text{ in lb} \]

Force on screws: \[ \frac{M}{9.9''} = \frac{2211.9}{9.9} = 223.4 \text{ lb} \]
The tension induced from the moment is shared by the top 8 screws (4 on 2 LM blocks). One screw would hold:

\[ \frac{223.4}{8} = 27.9 \text{ lbs axial} \]

Assuming only 4 out of the 16 screws hold the shear (load) due to misalignment:

\[ F_s = \frac{W_1 + W_2}{4} = \frac{276 + 6}{4} = 70.5 \text{ lb} \]

As for \( M_y = 8.78 \text{ mm}^2 = 0.0136 \text{ m}^2 \)

Bolt shears:

\[ \sigma = \frac{1}{A_s} \sqrt{F_s^2 + 3F_s^2} = \frac{1}{0.0136} \sqrt{27.9^2 + 3 \times 70.5^2} \]

\[ \sigma = 9.21 \text{ ksi} \]

\[ F_s = \frac{72.5 \text{ ksi}}{9.21 \text{ ksi}} = 7.9 \text{ based on ultimate} \]

\[ \geq 4 \text{ others} \]

Seismic Load:

\[ SD_{x1} = W_1 (C_{x1} \cdot (1p) \cdot (A_{x1}) \cdot (N_p) (40\%)) \]

\[ SD_{y1} = W_1 (C_{y1} \cdot (1p) \cdot (A_{y1}) \cdot (N_p) (100\%)) \]

\[ SD_{x2} = W_2 (C_{x2} \cdot (1p) \cdot (A_{x2}) \cdot (N_p) (40\%)) \]

\[ SD_{y2} = W_2 (C_{y2} \cdot (1p) \cdot (A_{y2}) \cdot (N_p) (100\%)) \]

\[ SD_{x1} = 172.2 \text{ lb} \]

\[ SD_{y1} = 430.56 \text{ lb} \]

\[ SD_{x2} = 6 (1)(1)(0.8)(1.2)(0.4) = 24.75 \text{ lb} \]

\[ SD_{y2} = 6 (1)(1)(1.0)(1.2)(1.0) = 9.36 \text{ lb} \]

Pull Pm = \( (SD_y + W_1) (8.0') + (SD_{y2} + W_2) (6.65') \) = 71.5 LB

**Note:** Eight screws sharing the load.
Pull from SD_x = \frac{SD_{x1} + SD_{x2} = 172.2 + 375}{16} = 11.0 \text{ lb}

16 screws sharing the load

Total seismic \( F_s = 71.5 + 11.0 = 82.5 \text{ lb} \)

\[ F_s = \frac{SD_{y1} + W_1 + SD_{y2} + W_2}{4} \text{ assume only 4 screws share the shear load} \]

\[ \sigma_s = \frac{1}{0.036} \sqrt{(82.5)^2 + 3(179.1)^2} = 23.6 \text{ ksi} \]

\[ F_s = \frac{72.6 \text{ ksi}}{23.6 \text{ ksi}} = 3.1 \]

\[ \sigma_s \geq 1.25 \text{ ok} \]

**Bolt Torque:**

\[ T = K_f \cdot D_n \cdot F_p \]

\[ F_p \geq 1.6(F_s) = 1.6(27.9) = 44.6 \text{ lb} \]

\[ F_p \geq 0.65(s_0)(A_n) = 72.5 E3(0.65)(0.0136) = 640.9 \text{ lb} \]

\[ K_f = 0.3 \text{ (cleanroom, high suction)} \]

\[ T = 0.3 \left( \frac{44}{25.4} \right)(350) = 16.5 \text{ in-lb} \]

**NOTE:** Thread pull out is included in the manufacturers rated load for the LM Blocks.
LM Block + Rail Rating:

THK model 4 SHS-15R (see attached manufacturer data sheets)

This model was selected because it is designed to have "4-way equal load rating".

Manufacturer Rating:

Static, \( C_0 = 24.2 \text{ kN} = 5,440 \text{ lb} \)

Dynamic, \( C = 9.8 \text{ kN} = 2,200 \text{ lb} \)

The 4 LM Blocks will resist the same moment + shear load that was used on the screws.

\[
V = \frac{W_1 + W_2}{4} = 70.5 \text{ lb}
\]

\[
H = \frac{M}{9.9'' \times \frac{1}{2}} = \frac{2211.9}{2(9.9)} = 111.7 \text{ lb}
\]

Total load per LM block: \( \sqrt{70.5^2 + 111.7^2} = 132 \text{ lb} \)

This is far from the 2,200 lb dynamic rated load.

Rail Screws

Each of the two rails have a single row of 8 M4 screws placed 2.36" apart.

The same moment load applies (assumes the LM block's weight would contribute an insignificant amount in the moment of this part).
Assume that 2 screws take the tension load for each LM block in tension. (4 total)

\[ F_a = \frac{M}{4(99)} = 55.9 \text{lb} \]

Assume only 4 screws take the shear load due to misalignment.

\[ F_s = \frac{W_1 + W_2 + Z}{4} = 71.0 \]

\[ \sigma = \frac{1}{0.0136} \sqrt{55.9^2 + 3 \times 71.0^2} = 9.87 \text{ksi} \]

\[ F_S = \frac{72.5}{9.87} = 7.3 \geq 4: \text{okay} \]

Seismic:
- Use same loads as seismic case for LM block screws:

\[ SD_x = 17.2 \text{ lb} \quad SD_y = 3.79 \text{ lb} \quad W_1 = 276 \text{ lb} \]
\[ SD_x = 430.52 \text{ lb} \quad SD_y = 9.36 \text{ lb} \quad W_2 = 610 \text{ lb} \]

Pull from:
\[ SD_y = \frac{(SD_y + W_2)(8.0) + (SD_x + W_1)(0.65)}{9.9 \times 4} = 14.3 \text{ lb} \]

Pull from:
\[ SD_x = \frac{SD_x + SD_x}{8} = 22.0 \text{ lb} \]

\[ F_{ax} = 14.3 + 22.0 = 11.65 \text{ lb} \]

\[ F_s = \frac{W_1 + SD_y + W_2 + SD_x}{4} = 179.1 \text{ lb} \]

\[ O_s = \frac{1}{0.0136} \sqrt{165^2 + 3(179.1)^2} = 25.8 \text{ ksi} \]

\[ F_S = \frac{72.5}{9.87} = 7.3 \geq 1.25: \text{ok} \]
Look at the thread in the aluminum stripping:

Seismic: \( F_a = 165 \text{ lb} \)

\[ A_t = \frac{\pi}{2} \times D_n \times L_e = \frac{\pi}{2} (0.067')(\frac{12\text{ mm}}{25.4\text{ in}}) = 0.117 \text{ in}^2 \]

\[ \tau = \frac{F_a}{A_t} = \frac{165\text{ lb}}{0.117\text{ in}^2} = 1.41 \text{ KSI} \]

\[ F_S = \frac{45}{65(1.41)} = \frac{18.4}{18.4} \geq 1.25 \text{ ok} \]

Do not need to look at static case!

**Rail Block + Threaded Rod**

The rail block is designed so that there will be minimal force exerted on it. The weight and moment from the cantilevered mast will be taken up mostly by the rail + rail block assemblies, because the rail block has slightly compressed Belleville washers to take up any misalignment.

There are no calculations needed for these pieces.

**NOTE:** There is a shield around the rail block + threaded rod to catch any particulate that may be created.
**Rail Plate Hinges**

\[ W_1 = 276 \]

\[ W_3 = 2 \times \text{Rail} + \text{LM Block} \times 4 + \text{Threaded Rod} + \text{Rail Plate} + W_z \]

\[ 1.94'' \text{ round to } 1'' = 2.5 \text{ lb} \]

\[ M = 9.6(W_1) + 1(W_3) = 2676.1 \text{ in lb} \]

Tension load on top two bolts:

\[ F_a = \frac{2676.1}{2(5.76)} = 232.3 \text{ lb} \]

Shear from weight:

\[ \frac{W_1 + W_3}{2} = \frac{151.3}{2} \text{ lb} \]

Assume only two bolts carry the load due to mass alignment.

There is a possibility that the mirror subassembly is not centered on the rail. In the LRU assembly, it does say to make sure it is centered, but there is a possibility it is not, which will create additional shear stress due to twist.

Only \( W_1 \) and the .7 lbs from the 2 LM mounts and the 4 LM Blocks will add to the twist moment:

\[ W_E = 276 + 7 = 283.1 \text{ lb} \]

\[ SD_E = 283.1(0.01)(0.01)(45)(45) = 4411.5 \text{ lb} \]
Shear from twist (assume two bolts due to misalignment)

\[ M_e = 283 \times 4'' = 1132 \text{ in-lb} \]

\[ r = \sqrt{\left(\frac{3.76}{2}\right)^2 + \left(\frac{0.94}{2}\right)^2} = 2.92 \text{ in} \]

\[ \tan^{-1}\left(\frac{3.76}{0.94}\right) = 80.7° \]

Shear twist force

\[ \frac{M_e}{2r} = \frac{1132}{2(2.92)} = 193.8 \text{ lb} \]

\[ F_s = \text{worst case vector build-up} \]

\[ F_s = \sqrt{(151.3)^2 + (193.8 \sin 80.7°)^2} \]

\[ = 264.47 \text{ lb} \]

There is also a small moment on the bolt where the hinge meets the yoke plate.

\[ M = 0.98(302.5) = 296.5 \text{ in-lb} \]

\[ \sigma = \frac{M_e}{I} \]

The bottom is free so the cross section for \( c = I \) will be taken to be from the bottom of the screws to the top of the hinge.

\[ C = \frac{625}{2} + 134° = .22'' \]

\[ I = \frac{1}{12}bh^3 \text{ (because beam is actually the hinge) we are just finding flexural stiffness) } \]

\[ I = \frac{1}{12}(1.57)(0.45)^3 \approx 0.119 \text{ in}^4 \]
\[ \sigma = \frac{M_L}{I} \pm \frac{F_S}{A_s} = \frac{298.5 \text{ lb} (12^2)}{0.0119 \text{ in}^4} + \frac{720.5}{0.0567} \]

\[ \sigma = 9.58 \text{ ksi} \quad A_s = 0.0567 \text{ in}^2 \]

For M8 bolt

\[ t = \frac{E_s}{A_s} = \frac{264.47}{0.0567} = 4.66 \text{ ksi} \]

\[ \sigma_{max} = \sqrt{\sigma^2 + 3t^2} = \sqrt{9.58^2 + 3(4.66)^2} = 12.53 \text{ ksi} \]

\[ FS = \frac{72.5}{12.53} \geq \frac{15.8}{4} \geq 4 \text{, ok} \]

Note: This calculation was conservative because it did not take into account that the hinge is sealed in a bolt of the yoke plate. It is designed to have a small interference fit which will take much of the shear load off the bolts.

Thread slippage:

\[ M_8 \times 20 \]

\[ \frac{20 - 6.88 \text{ mm}}{25.4} = .5 \text{" length of engagement} \]

\[ A_t = \frac{\pi D t L_e}{2} = \frac{\pi (8 \text{ mm})(.5)}{25.4} = .247 \]

\[ t = \frac{E_s}{A_t} = \frac{232.5}{.247} = 0.94 \text{ ksi} \]

\[ FS = \frac{615}{3(0.94)} = 27.0 \]

\[ t \text{ cause in shear} \]

2 \text{, ok}
Bolt Torque:

\[ T = K D_0 F_p \]

\[ F_p \leq 1.6 F_a = 1.6 \left( \frac{282.3}{\sqrt{3}} \right) = 371.5 \text{ lb} \]

\[ F_p \leq 0.65 (A_s)(S_{ut}) = 0.65 \left( \frac{1}{0.8567} \right) \left( \frac{72.563}{2} \right) = 2.672 \text{ lb} \]

Choose: \( F_p = 1600 \text{ lb} \)

\[ T = 0.3 \left( \frac{8/23.4}{1800} \right) = 151 \text{ in lb} \]

Seismic Rail Plate Bolts:

\[ SD_x = W(Co)(N_r)(A_{cc})(S_{ut}) \]

\[ SD_y = W(Co)(1)(A_{cc})(N_r)(L_i) \]

\[ SD_{x1} = 172.2 \text{ lb} \quad SD_{x2} = 16.5 \text{ lb} \]

\[ SD_{y1} = 480.6 \text{ lb} \quad SD_{y2} = 41.3 \text{ lb} \quad SD_{y} (P95/2) = 441.5 \text{ lb} \]

Pull FROM \( SD_y = \frac{(SD_{y1} + W_i)(.6)}{(2)(.76)} \)

\[ = 594.73\text{ lb} \]

Pull FROM \( SD_x = \frac{SD_{x1} + SD_{x2}}{4} = 47.2 \]

\[ F_s = 594.7 + 47.2 = 641.9 \text{ lb} \]

Shear FROM weight = \( SD_{x1} + W_i + SD_{y2} \cdot W_i = 387.2 \text{ lb} \)

Shear FROM twist \( \left( \frac{(W_i + SD_{y1})^4}{2} \right) \left( \frac{283 + 441.5}{2} \right) = 496.2 \text{ lb} \)

\[ F_s: \sqrt{(387.2 + 496.2)(\cos 80.7)^2 + (496.2 \sin 80.7)^2} = 676.9 \text{ lb} \]

Bending moment = \( 0 \)

(next page)
\[ M = 0.98 \left( 392.6 + 5D_{y1} + 5D_{y2} \right) = 900.0 \text{ mIbf} \]

Same \( C + I \) from before

\[ S = \frac{Mc}{I} + \frac{F_a}{A_s} = \frac{900.0(0.22)}{0.019} + \frac{641.9}{0.0567} \]

\[ = 27.96 \text{ ksi} \]

\[ \epsilon = \frac{F_a}{A_s} = \frac{676.9}{0.0567} = 11.94 \text{ ksi} \]

\[ C_{v02} = \sqrt{27.96^2 + 3(11.94)^2} = 34.78 \text{ ksi} \]

\[ FS = \frac{725}{34.78} = \frac{2.1}{1.25} \geq 1.25 \text{ ok} \]

NOTE: There is a conteracting moment on the pin that was not taken into account when calculating the bending moment on the hinge. If this was accounted for, the moment on the Schely's would have increased, making the safety factor higher.
Tilt Plate, Pins

- All pins are A2-70 with \( \frac{3}{4} '' \) DIA, \( V_{\text{shear}} = 10.5 \text{ kips} \), and are pressed into the tilt plate.

- Pins 1 and 3 are used with the yaw plate.

- Pins 2 and 4 are used with the pitch plate.

Pins 1 and 3 are worst case because pin 1 has a significant bending stress that the others don't.

\[
W_1 = 276 \text{ lb} \quad \text{from pg 13}
\]

\[
W_3 = 26.5 \text{ lb}
\]

\[
W = 276 \times (10.58') + 26.5 \times (1.92') = 2971.9 \text{ lb}
\]

\[
F_m = \frac{2971.9}{5.13'} = 579.1 \text{ lb}
\]

There is also shear from the twist moment from the small chance that in mirror is off-center.

\[
M_t = 1132 \text{ in lb (from pg 13)}
\]

\[
F_t = \frac{M_t}{5.13'} = 220.66 \text{ lb}
\]
Top 8 Pin:

Total shear force, \( F_s = \sqrt{F_m^2 + F_t^2} \)

\[ F_s = \sqrt{579.1^2 + 220.66^2} = 579.1 \text{ lb} \]

\[ \tau = \frac{F_t}{A} \]

\[ A = \pi r^2 = \pi \left( \frac{1}{4} \right)^2 = 0.196 \text{ in}^2 \]

\[ \tau = \frac{4.2 \times 619.7}{0.196} = 4.2 \text{ ksi} \]

There is no axial force, but there is a bending moment on the top pin.

There is no axial force, but there is a bending moment on the top pin.

\[ M = 302.6 \times 0.98 \times 0.298.5 \text{ in lb} \]

\[ \sigma = \frac{Mc}{I} \]

\[ c = \frac{1}{4} \]

\[ I = \frac{\pi}{64} (.25)^4 = .00307 \text{ in}^4 \]

\[ \sigma = \frac{298.5 \times .25}{.00307} = 24.1 \text{ ksi} \]
\[
\sigma_{vms} = \sqrt{\sigma^2 + 3 \tau^2} = \sqrt{24.16^2 + 3(4.21)^2} = 29.23 \text{ksi}
\]

Stress for AZ-70 pin = 101.6 ksi

\[
FS = \frac{101.5}{29.23} = 4.0 \quad \geq 4 \quad \text{so OK}
\]

Note: this was a conservative calculation because some of the bending moment in the top pin would be taken up by the friction on the bottom pin.

**Seismic**

There has been no added weight so the seismic values from the sail plate bolts can be used.

- SD_{x1} = 172.8 lb
- SD_{y1} = 430.6 lb
- SD_{x2} = 441.6 lb
- SD_{y2} = 41.3 lb

Shear from moment:

\[
F_M = \frac{(SD_{y1} + W)(10.58^\circ) + (SD_{y2} + W)(1.92^\circ)}{5.13} = 1482.6 \text{ lb}
\]

Shear from stress:

\[
F_s = \frac{W + SD_{y1}}{5.13} = 5687 \text{ lb}
\]
Total Shear:

\[
Fs = \frac{(F_{w} + F_{s})^2 + (F_{t})^2}{\sqrt{(1482.6 + 94.35)^2 + (568.7)^2}}
\]

\[
Fs = 1676.4 \text{ lb}
\]

\[
\tau = \frac{4Fs}{3h} = \frac{4(1676.4)}{3 \cdot 0.196} = 11.4 \text{ ksi}
\]

Bending moment:

\[
M = (w_{1}w_{2} + 0.05w_{3})(0.98) = 759.01 \text{ in lb}
\]

\[
\sigma = \frac{Mc}{I} = \frac{759.01(1.25)}{0.00307} = 61.81 \text{ ksi}
\]

\[
\sigma_{max} = \sqrt{(61.81)^2 + 3(11.4)^2} = 64.89 \text{ ksi}
\]

\[
FS = \frac{101.5}{64.89} = 1.6 \geq 1.25 \text{ : OK}
\]

Note: This really includes two rare events: 1, an earthquake and 2, the mirror being off-center by 4".

Pitch Plate:

*The Rail Plate can rotate ±20° relative to the rest of the NOID End Effector + NOID arm. The worst case stresses occur when the Rail Plate is rotated all 20° because it adds to the twist moment. That places a shear stress on the rest of the assembly's fasteners. For the remainder of the calculations (except where noted), calculations will assume the 20° rotated position.*
Looking at Pitch Plate bolts

\[ M_b = W_1 \cdot (9.62) + W_3 \cdot (1.97 + 1.75) + W_4 \cdot (1.75) \]
\[ = 2730.7 \text{ in lb} \]

\[ M_t = (w_1)(3.73 + 3.76) + W_3 \cdot 0.72 \]
\[ = 2086.3 \text{ in lb} \]

\( M_t \) is taken only by the 4 screws

\( M_b \) is shared between the screws and the threaded rod.
\[ F_a = \frac{M_b}{4(z_d)} = \frac{2730.7}{4(6.4)} = 126.4 \text{ lb} \]

2 screw

\[ F_s = \frac{\sum W}{2} = \frac{276 + 26.5 + 3}{2} = 152.8 \text{ lb} \]

assumes 2 screws share load due to misalignment

\[ \theta = \tan^{-1}\left(\frac{.94}{5.0}\right) = 10.6^\circ \]

\[ r = \sqrt{\left(\frac{.94}{2}\right)^2 + \left(\frac{5.0}{2}\right)^2} = 2.54' \]

\[ F_{s\text{twist}} = \frac{M_b}{2(2.54)} = \frac{2086.3}{2(2.54)} = 410.08 \text{ lb} \]

assumes 2 screws share load

Total Shear:

\[ F_s = \sqrt{(F_{s\text{weight}} + F_{s\text{twist}} \cos \theta)^2 + (F_{s\text{twist}} \sin \theta)^2} \]

\[ = \sqrt{(152.8 + 403.1)^2 + (75.4)^2} \]

\[ = 560.97 \text{ lb} \]

\[ \sigma = \frac{1}{A} \sqrt{F_a^2 + 3F_s^2} = \frac{1}{0.0567} \sqrt{(126.4)^2 + (560.97)^2 \times 3} \]

\[ = 17.28 \text{ ksi} \]
\[
FS = \frac{72.5}{17.28} = \sqrt[4]{4.2} \geq 4 \; \therefore \text{OK}
\]

Note: Many conservative assumptions were made in this calculation:
1. Assembly at max (20\(^\circ\)) angle
2. MSA offset center on rails 4"
3. Only 2 out of the 4 screws take the shear load for weight + twist
4. A 2-50 set was used instead of the A 2-70 set value.

Thread stripping: similar to other hinges which had a very high
Bolt Torque: use same torque as M8 bolts in the rail plate hinges. (pg 16)

Seismic

\[
\begin{align*}
SD_x &= W (CP)(IP)(ACC)(NP) (96\%) \\
SD_y &= W (CP)(IP)(ACC)(NP) (100\%) \\
SD_{dx} &= 172.2 \\
SD_{dy} &= 430.6 \\
SD_{dx} &= 16.9 \\
SD_{dy} &= 41.3 \\
SD_{dx} &= 1.9 \\
SD_{dy} &= 4.7
\end{align*}
\]

\[
\begin{align*}
M_{t_x} &= (SD_y + W_y)(9.23 + 3.76) + (SD_y + W_z)(1.97 + 7.5) + (SD_y + W_y)(7.25) \\
&= 5341.3 \text{ in-lb} \\
M_{b_y} &= (SD_y + W_y)(9.62) + (SD_y + W_z)(19.71 + 7.57) + (SD_y + W_y)(7.25) \\
&= 6987.7 \text{ in-lb}
\end{align*}
\]

Because the rail assembly is free to rotate about Y, it is assumed that SD\(_x\) puts an even axial pull across the four bolts.
\[ Fa = F_{a_y} + F_{sd, r} \]
\[ F_{sd, r} = \frac{\sum SD_r}{4} = \frac{172.2 + 16.5 + 1.9}{4} = 47.65 \text{ lb} \]
\[ Fa_b = \frac{M_b}{5.4'(4)} = \frac{6987.7}{4(5.4')} = 323.5 \text{ lb} \]
\[ F_c = 47.65 + 323.5 = 371.15 \text{ lb} \]
\[ F_s = \text{resultant of } F_{\text{weight}} + F_{\text{twist}} \]
\[ F_{\text{weight}} = \frac{\sum W + \sum SD_y}{2} = 391.05 \text{ lb} \]
\[ F_{\text{twist}} = \frac{M_t}{2.54} = \frac{5341.3}{2(2.54)} = 1051.44 \text{ lb} \]
\[ F_s = \sqrt{(F_{\text{weight}} + F_{\text{twist}} \cos \theta)^2 + (F_{\text{twist}} \sin \theta)^2} \quad \theta = 10.6^\circ \]
\[ F_s = \sqrt{(391.05 + 1033.5)^2 + (193.4)^2} \]
\[ F_s = 1437.6 \text{ lb} \]
\[ \sigma = \frac{1}{A_t} \sqrt{Fa^2 + 3F_s^2} = \frac{1}{0.0567} \sqrt{(371.15)^2 + 3(14376)^2} \]
\[ \sigma = 44.40 \text{ ksi} \]
\[ FS = \frac{72.5}{44.40} = 1.63 \geq 1.25 \quad \text{OK} \]

Note: Some stock up & conservative assumption applied here (see pg 21)
Threaded Rod Through Pitch Plate

Worst case for the threaded rod is when there is no rotation in yaw.

Looking for thread shear:

\[ M_b = W_1 (10.9 + .75) + W_3 (2.1 + .75) + W_4 (1.75) \]
\[ = 3293.2 \text{ in lb} \]

\[ F = \frac{M_b}{5.4} = \frac{3293.2}{5.4} = 609.85 \text{ lb} \]

Brass is the weakest material with \( S_{ut} = 55,000 \text{ psi} \) (range of 55-75 ksi given by manufacturer).

\[ L_e = \frac{1}{2}, \quad D_n = \frac{3}{8}, \quad TPI = 24 \text{ (fine)} \]

\[ A_0 = \frac{\pi}{4} \left( D_n \right) \left( L_e \right) = \frac{\pi}{4} \left( \frac{3}{8} \right) \left( \frac{1}{2} \right) = 0.26 \text{ in}^2 \]

\[ \tau = \frac{F}{A_0} = \frac{609.85}{0.26} = 2350 \text{ ksi} \]

\[ FS = \frac{S_{ut}}{\sqrt{3} \tau} = \frac{55.0 \text{ ksi}}{\sqrt{3} (2350 \text{ ksi})} = 13.5 \geq 4 : \text{ ok} \]

Seismic:

All seismic values from pg 24 still apply!
$$M_{\text{basic}} = (SD_x + W)(10.9 + 7.5) + (SD_y + W)(2.1 + 7.5) + (SD_y + W)(.75)$$
$$= 8430.9 \text{ inlb}$$

$$F_{sd_x} = \frac{8430.9}{5.4} = 1561.3 \text{ lb}$$

$$F_{sd_y} = SD_x + SD_y + SD_y = 95.3 \text{ lb}$$

2 assume other half is taken by pins.

Total F = 1561.3 + 95.3 = 1656.6 lb

$$\tau = \frac{F}{A_s} = \frac{1656.6}{1.26} = 6.37 \text{ ksi}$$

$$FS = \frac{S_{w + brass}}{\sqrt{3}} = \frac{55.0}{\sqrt{3}} = \left[\frac{55.0}{6.37}\right] = 8.8$$

Contact Stress:

(From Roark's Formulas for Stress + Strain)

\[ C = \left(1 - \frac{V^2}{E_1}\right)^{1/2} = 2.027 \times 10^{-8} \]

Tungsten Carbide

\[ \frac{k}{\alpha} = \frac{D_1 D_2}{D_1 D_2 - \frac{D_1}{4} - \frac{D_2}{4}} = 1.75 \]

\[ \sigma = \frac{0.721}{\sqrt{160.85 \times 1.75 \times 2.027 \times 10^{-8}}} \]

\[ \alpha = 0.02 \text{ in}^2 \]

\[ \delta_1 = \frac{790}{7.216 \times 10^{-5}} \]

\[ SF = \frac{7.90 \times 10^{-5}}{7.215 \text{ psi}} = 1.1 \]

This is not critical joint and yield will only decrease stress! 

\[ \begin{align*}
\tau_1 &= \frac{55.0}{1.26} = 43.7 \\
\tau_2 &= \frac{55.0}{1.26} = 43.7 \\
\tau_3 &= \frac{55.0}{1.26} = 43.7 \\
\end{align*} \]
Force to turn the crank:

\[ F \rightarrow \begin{array}{c}
\text{5/8"-24 Lubricated Brass} \\
\text{3/8"-24 Keensert}
\end{array} \text{ G T I 1.75}

T = K(D_n)(F)

Where \( F = 609.85 \text{ lb} \)
\( D_n = 3/8" \)

\[ K = 0.10 \quad \text{(Estimated value from the K values in the HE Design Standards. It's low because lubricated brass and SS7 have a low coefficient of friction)} \]

\[ T = 0.10 \times 3/8" \times 609.85 \text{ lb} = 22.87 \text{ in lb} \]

\[ F_{\text{handle}} = \frac{T}{1.75} = \frac{13.2 \text{ lb}}{1.75} \]

This is doable considering this is the worst case scenario. Most mirror assemblies weigh \( 3/8" \) the weight used in the calculation, and the friction is probably less than what was estimated. If there is a problem when it is assembled, a longer handle can be used to decrease the force required to turn the handle.

Seismic does not have to be calculated because no one will have to turn the handle at such a time.
Shaft

6 x M8 (Ag PLD SST)

φ 3.5" 1.04

W₁ = 276 (MSA)
W₂ = 26.5 lb
W₃ = 26.7 lb
(W₄ + pitch plate, + crank)

Mt = 2086.3 (from pg 22)
Mₖ = W₁(10.37") + W₂(19.71 lbf) + W₃(1.8")
= 2929.2 in lbf
\[
F_a = \frac{M_b}{z(3.04)} = \frac{2929.2}{2(3.04)} = 356.1 \text{ lb}
\]

**Weight:***

\[
F_{\text{weight}} = \frac{276 + 26.5 + 6.7}{3} = 103.07 \text{ lb}
\]

**Twist:**

\[
r - \frac{d}{2} = \frac{3.5}{2} = 1.75''
\]

\[
F_{\text{twist}} = \frac{M_b}{3(1.75)} = \frac{2086.3}{3(1.75)} = 397.4 \text{ lb}
\]

**Worst Case:**

\[
F_3 = F_{\text{weight}} + F_{\text{twist}} = 103.07 + 397.4 = 500.46 \text{ lb}
\]

\[
A_t = 0.0567 \text{ for } M^2
\]

\[
\sigma = \frac{1}{A_t} \sqrt{F_a^2 + 3F_3^2} = \frac{1}{0.0567} \sqrt{(356.1)^2 + 3(500.46)^2}
\]

\[
= 16.53 \text{ ksi}
\]

\[
FS = \frac{72.5}{16.53} = \left[ \frac{4.4}{4} \right] \geq 4 \text{ : OK}
\]

**Pull Out:**

For worst case is when the rail plate has 0° rotation
\( M_b = W_1 (12.3) + W_3 (3.5) + W_5 (.8) = 4722.9 \text{ in} \cdot \text{lb} \)

\( F_a = \frac{M_b}{2(304')} = 776.8 \text{ lb} / \text{screw} \)

Screws: M8 x 25 into 304 stainless steel (\( 3_y = 35 \text{ ksi} \))

\( A_s = \frac{\pi}{2} \left( D_n - D_p \right)^2 \cdot L_c \)

\( = \frac{\pi}{2} \left( 3.15 - .05 \right) \cdot 5 = .208 \text{ in}^2 \)

\( T = \frac{F_a}{A_s} = \frac{776.8}{.208} = 373 \text{ ksi} \)

\( F_S = \frac{35 \text{ ksi}}{\sqrt{3} \cdot (3.73)} = \frac{5.4}{\sqrt{3}} \geq 3 \text{ OK} \)

Torque:

\( F_p \geq 1.6 (F_a) = 1.6 \cdot 776.8 = 1242.9 \text{ lb} \)

\( F_p \leq 0.65 (A_t)(S_u) = .65 \cdot (\pi r^2) \cdot (72.6 \text{ ksi}) = 2672.0 \)
choose $F_p = 2000$ lb

$T = K \cdot D_n \cdot F_p$

$= 0.25 \cdot 0.315 \cdot 2000 = 157.5 \rightarrow 160$ in lb

Seismic

$SD_y = W \cdot GP \cdot (1-P) \cdot (Acc)(N_p)$

$SD_x = 40\% (SD_y)$

$SD_{nx} = 172.2$

$SD_{sy} = 430.6$

$SD_{sx} = 16.5$

$SD_{sy} = 41.3$

$SD_{sx} = 4.2$

$SD_{sy} = 10.6$

$M_b = 5341.3$ (some as previous calc on pg 24)

$M_b = (SD_{ny} + W \cdot 10.27 + (SD_{sy} + W_s)(1.97 + 1.4) + (SD_{sy} + W_s)(1.8))$

$= 7499.0$ in lb

$F_a = F_{ax} + F_{asx}$

$F_{asx} = \frac{\Sigma SD_x}{6 \cdot 6\text{screws}} = 32.15$ lb

$F_{ax} = \frac{M_b}{2(3.04)} = \frac{7499}{2(3.04)} = 1233.4$ lb

$F_a = 32.15 + 1233.4 = 1265.5$ lb

$F_s = F_{ax} + F_s \text{ weight}$

$F_s \text{ weight} = \frac{\Sigma W + \Sigma SD_y}{3}$

$\text{assumes 3 out of the 6 screws}$

$= 26.39$ lb

Share the load
\[ F_s_{axial} = \frac{M_b}{3 \ r} = \frac{5341.3}{3 \ (1.75)} = 1017.4 \]

\[ F_s = 263.9 + 1617.4 = 1281.3 \]

\[ \sigma = \frac{1}{0.056} \sqrt{1265.5^2 + 3(1281.3)^2} = 45.6 \ \text{ksi} \]

\[ F_s = \frac{72.5}{45.6} = 1.6 \geq 1.25 : \text{ok} \]

**Stresses in the Shaft and Bearings**

\[ M_b = 2929.2 \]
\[ M_t = 2086.3 \]
\[ \Sigma W = 276 + 26.5 + 6.7 = 309.2 \ \text{lb} \]

\[ \Sigma F_y = 0 \]
\[ \Sigma W + R_z \cdot R_1 \quad \text{(1)} \]

\[ \Sigma M_2 = 0 \text{ at origin} \]

\[ M_b + 1.34(R_1) = R_z (1.34 + 3.16) \]
\[ 2929.2 + 1.34R_1 = 4.5R_z \quad \text{(2)} \]

Substitute eqn (1) into eqn (2):

\[ 2929.2 + 1.34(309.2 + R_z) = R_z (4.5) \]
\[ 3341.3.5 = 316R_z \]

\[ R_z = 1058.1 \ \text{lb} \]
$R_1 = 1367.3 \text{ lb}$

**Bearing Rating (from Timken.com)**

<table>
<thead>
<tr>
<th>Type</th>
<th>Rating</th>
</tr>
</thead>
<tbody>
<tr>
<td>#5261622</td>
<td>+ 23620/23690</td>
</tr>
</tbody>
</table>

Dynamic Radial

<table>
<thead>
<tr>
<th>Rating</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>8,900</td>
<td>4,800</td>
</tr>
</tbody>
</table>

Note: There is very little thrust on these bearings, so there is no need to look at it. The rated thrust load for the small bearing is 3040 lb.

$$FS = \frac{8,900}{1367.3} = 6.51 + \frac{4,800\text{ lb}}{1058\text{ lb}} = 4.54$$

**Shaft:**

- $M_b = 2929.2 \text{ in lb}$
- $2W = 309.2 \text{ lb}$
- $R_1 = 1367.3 \text{ lb}$
- $R_2 = 1058.1 \text{ lb}$

In addition to shear and the bending moment there is also a twist, $Mt$, the shaft is carrying.
A, B, and C are the critical stress areas of the shaft. The stress will be calculated at all three.

At Point A:

\[ M_c = 2929.2 \text{ in lb} + 309.2(1.75) = 3611 \text{ in lb} \]

\[ V = 309.2 \text{ lb} \]

\[ M_t = 2086.3 \text{ in lb} \]

For DIA = 1.625'' ~

\[ I = \frac{\pi}{4} r^4 = 0.342 \text{ in}^4 \]

\[ A = \pi \frac{d^2}{4} = 2.07 \text{ in}^2 \]

\[ C = \frac{h}{2} d = 0.813 \text{ in} \]

\[ J = \frac{1}{2} C^2 = 1.689 \text{ in}^4 \]

\[ \sigma_a = \frac{M_c}{I} = \frac{3611 \times (0.813)}{0.342} = 8414 \text{ ksi} \]

The decrease in diameter causes a stress concentration

\[ \frac{D}{d} = \frac{4.5}{1.625} = 2.77 \]

\[ \frac{r}{d} = \frac{0.079}{1.625} = 0.049 \]

\[ K = 2.2 \]

\[ \sigma = 8414 (2.2) = 18,577 \text{ ksi} \]

Shear →

\[ \tau_1 \quad \tau_2 \]
\( \tau_v = \frac{V}{A} = \frac{309.2}{2.07} = 149.41 \text{ psi} \)

\( \tau_t = \frac{M_t c}{J} = \frac{2086.3 \text{ in lb} \cdot \text{(8.13 in)}}{0.684 \text{ in}^4} = 2.47 \text{ ksi} \)

From pg 283 of the Machinery's Handbook 25th edition:

\( K = 1.8 \)

\( \tau_b = 2.47 \cdot 1.8 = 4.45 \text{ ksi} \)

\( \tau = \tau_v + \tau_b = 4.60 \text{ ksi} \)

\( \sigma_{vms} = \sqrt{\sigma_v^2 + 3 \tau^2} = \sqrt{18.57^2 + 3(4.6)^2} = 20.2 \text{ ksi} \)

Sut SST304 = 85 ksi

\( SF = \frac{85}{20.2} = 4.2 \geq 4.0 \text{ ok} \)

NOTE: if the assumptions made that there are no stress concentrations because SST304 is ductile + the SF is calculated based on \( \sigma_{vms} = 25 \text{ ksi} \)

\( SF = \frac{185}{3.7} \leq 50 \text{ still good} \)

At Point B:

\( M_B = 3343.5 \text{ in lb} \)

\( V_B = 1367.3 \text{ lb} \)

\( M_t = 2086.3 \)

\( DIV = 1.625 \)

Point A:

\( V_A = 1.97 \text{ in lb} \)

\( M_c = 1367.3 \text{ in lb} \cdot \text{in} \)

\( \sigma = \frac{M_c}{I} = \frac{3343.5 \cdot 0.813}{1.342} = 9.45 \text{ ksi} \)

\( \tau = \frac{V_A}{A} + \frac{M_c}{J} = \frac{1367.3}{2.07} \cdot \frac{2086.3 \cdot 0.813}{0.684} = 0.66 \text{ ksi} \)

\( \sigma_{vms} = \frac{7.95^2 + 3.66^2}{4} = 8.03 \text{ ksi} \)
At Point C:

\[ M_c = 3343.5 - 1058.1(2.12^2 - 1.34) = 2518.2 \text{ in lb} \]

\[ V_c = 1058.1 \text{ lb} \]

\[ M_t = 2686.3 \text{ in lb} \]

\[ DIA = 1.375'' \Rightarrow A = \frac{\pi}{4} (1.375)^2 = 1.48 \text{ in}^2 \]

\[ I = \frac{\pi}{4} (.688)^4 = 0.175 \text{ in}^4 \]

\[ C = 0.688 \text{ in} \]

\[ J = 0.35 \text{ in}^4 \]

\[ \sigma' = \frac{M_c C}{I} = \frac{2518.2 (.688)}{1.175} = 9.90 \text{ ksi} \]

\[ \frac{P}{d} = \frac{1.625}{1.375} = 1.18 \quad \frac{r}{d} = \frac{0.125}{1.375} = 0.091 \]

\[ K = 1.65 \]

\[ \sigma' = 1.65(9.9) = 16.34 \text{ ksi} \]

\[ \tau_v = \frac{V_c}{A} = \frac{1058.1}{1.48} = 715 \text{ ksi} \]

\[ \tau_{t_0} = \frac{M_t C}{J} = \frac{2686.3 (.688)}{0.35} = 4.10 \text{ ksi} \]

\[ K = 1.35 \]

\[ \tau_t = 1.35(4.10) = 5.54 \text{ ksi} \]
\[ \tau = \tau_v + \tau_e = 0.715 + 5.54 = 6.25 \text{ ksi} \]

\[ \sigma_{\text{max}} = \sqrt{16.34^2 + 3(6.25)^2} = 19.6 \text{ ksi} \]

\[ \text{SF} = \frac{85}{19.6} = \frac{4.3}{2.9} \geq 4 \cdot \text{OK} \]

**Seismic**

\[ M_b = 7499 \text{ in lb} \]

\[ M_{b_2} = 5341.3 \text{ in lb} \]

Weight = \( \sum W_i \times S_i = 791.7 \text{ lb} \)

Any \( S_Dx \) will be taken by the bearings not carried in the shaft.

\[ \sum F_y = 0, \quad 791.7 + R_b = R_a \quad \text{(1)} \]

\[ \sum M_b + \text{ougen} = 0 \]

\[ M_b + R_a(1.34) = R_b(134 + 3.16) \]

7499 + 1.34R_a = 4.5R_b \quad \text{(2)}

Substitute eqn (1) into (2)

7499 + 1.84(791.7 + R_b) = 4.5R_b

\[ R_b = 2708.8 \text{ lb} \]

\[ R_a = 3500.5 \text{ lb} \]

Shear + Moment Diagram: on next page.
Dia = 1.625"
(I, A, C, + J from Pg 34)

At pt. A:

\[ M = 8092.8 \text{ in lb} \]
\[ V = 791.7 \text{ in lb} \]
\[ M_{t} = 5841.3 \text{ in lb} \]

\[ \sigma_{o} = \frac{M_{t}}{I} = \frac{8092.8 \times 0.813}{342.7} = 19.24 \text{ ksi} \]

\[ \sigma = K \sigma_{o} = 2.2(19.24) = 42.32 \text{ ksi} \]

\[ \tau_{v} = \frac{V}{A} = \frac{791.7}{3.67} = 382.5 \text{ psi} \]

\[ \tau_{t,o} = \frac{M_{t} c}{J} = \frac{5341.3 \times 0.813}{0.684} = 6.35 \text{ ksi} \]

\[ \tau = K \tau_{o} = 1.8(6.35) = 11.43 \text{ ksi} \]

\[ \tau = \tau_{v} + \tau_{t} = 382.5 + 11.43 = 393.93 \text{ ksi} \]

\[ \sigma_{ums} = \sqrt{42.32^2 + 3(11.81)^2} = 47.00 \text{ ksi} \]

\[ SF = \frac{85}{470} = \frac{1.8}{1.125} \geq 1.25: \text{ OK} \]

**NOTE:** If no assumption is made that there are no stress concentrations because SS1304 is a ductile material, the factor of safety as calculated based on yield (\( S_y = 35 \text{ ksi} \)) is \( SF = \frac{1.56}{1} \geq 1.5: \text{ OK} \)
Seismic (pt. B) → not needed because pts. A & C had higher stresses in the static case.

Pt. C (Seismic):

\[ M_c = 8559.9 \cdot 2708.8 \left( \frac{212}{2} - 1.34 \right) = 6447 \]

\[ V_c = 2708.8 \]

\[ \sigma_0 = \frac{M_c \cdot c}{I} = \frac{6447 \times 6.88}{6175} = 25.35 \]

\[ \sigma = 1.65 \times 25.35 = 41.8 \text{ ksi} \]

\[ T_v = \frac{V_c}{A_c} = \frac{2708.8}{1.48} = 1.83 \text{ ksi} \]

\[ T_{k_0} = \frac{5341.3 \times 6.88}{0.35} = 10.5 \text{ ksi} \]

\[ T_b = 1.35 \times 10.5 = 14.17 \text{ ksi} \]

\[ T = T_v + T_{k_0} = 14.18 \]

\[ \sigma_{rms} = \sqrt{41.8^2 + 3 \times (14.18)^2} = 49.03 \]

\[ F_S = \frac{85}{49.03} = \left[ \frac{1.73}{2.125} \right] \geq 1.0 \text{ because it's excessive} \]

Assuming no concentration factor, the stress is:

\[ \sigma_{rms} = \sqrt{25.35^2 + (12.53)^2} = 33.15 \text{ ksi} \]

\[ F_S = \frac{35}{33.15} = \left[ \frac{1.07}{1.0} \right] \geq 1.0 \text{ based on yield stress} \]
The key will take a shear load from the twist moment.

\[ M_t = 2086.3 \text{ in} \cdot \text{lb} \]

\[ F = \frac{M_t}{(1.3)\pi/2} = \frac{2086.3}{1.375/2} = 3034.6 \text{ lb} \]

\[ A = 0.281 \text{ in}^2 \]

\[ t = \frac{F}{A} = \frac{3034.6}{0.281} = 10.79 \text{ ksi} \]

\[ FS = \frac{90 \text{ ksi}}{\sqrt{3} (10.79)} = \frac{90 \text{ ksi}}{20.35 \text{ ksi}} = 4.8 \geq 4 \text{ ksi} \]

Seismic

\[ M_{ts} = 5341.3 \text{ in} \cdot \text{lb} \]

\[ F = \frac{M_{ts}}{(1.3)\pi/2} = 7769.2 \text{ lb} \]

\[ t = \frac{7769.2}{0.281} = 27.65 \text{ ksi} \]

\[ FS = \frac{90 \text{ ksi}}{\sqrt{3} (27.65)} = 1.9 \geq 1.25 \text{ ksi} \]

Force Ring

The force ring and its components take only the twist moment.
Looking for shear in the pin:

\[ F = \frac{M_t}{2} \]

\[ M_t = 2086.3 \text{ in lb} \]

\[ F = 1043.2 \text{ lb} \]

Pin: \( d = \frac{3}{8} \) in

\[ A = \frac{\pi d^2}{4} = 0.11 \text{ in}^2 \]

\[ \tau = \frac{4F}{3A} \geq \frac{1043.2}{3(0.110)} = 32.37 \text{ ksi} \]

\[ FS = \frac{101.5 \text{ ksi}}{32.37} = \frac{1.81}{4} \geq 4 \text{ ak} \]

Seismic:

\[ M_s = 5341.3 \]

\[ F = \frac{5341.3}{2} = 2670.65 \text{ lb} \]

\[ \tau = \frac{4(2670.65)}{3(0.110)} = 32.37 \text{ ksi} \]

\[ FS = \frac{101.5 \text{ ksi}}{32.37} = \frac{1.81}{4} \geq 4 \text{ ak} \]

Stressed in the force ring:

Shear

\[ F = 1043.2 \text{ lb} \]
\[ \tau = \frac{3F}{2A} = \frac{3}{2} \frac{1043.2}{(254 \times 1.75)} = 7.9 \text{ ksi} \]

*Rectangular section *

F.S. = \[ \frac{40}{\sqrt[3]{7.9}} \]

*Based on yield, it should be greater or equal to 3 *

\[ \sigma = \left[ \frac{2.92}{2} \right] \]

\[ \sigma = \left[ \frac{2.92}{2} \right] \]

*Based on yield, it should be greater or equal to 3 *

\[ \sigma = \frac{621.6}{0.0625} = 8.35 \text{ ksi} \]

\[ \sigma_{ms} = \sqrt{8.35^2 + 3(2.09)^2} \]

\[ \sigma_{ms} = 9.10 \text{ ksi} \]

\[ \text{F.S.} = \frac{40}{91} = \left[ \frac{4.41}{2} \right] \geq 3 \]

*Based on yield, it should be greater or equal to 3.*
Seismic

Shear at top:

\[ F = 2670.65 \text{ (Pg 42, Seismic)} \]

\[ \tau = \frac{3}{2} \left( \frac{2670.65}{0.264 \times 1.75} \right) = 20.23 \]

\[ \text{FS.} = \frac{40}{\sqrt[3]{20.23}} = \frac{40}{1.14} \geq 1 \quad \text{OK} \]

based on yield

Stress at the base:

Same seismic F: \[ F = 2670.65 \text{ lb} \]

\[ \tau = \frac{3}{2} \left( \frac{2670.65}{0.75 \times 1'} \right) = 5.34 \text{ ksi} \]

\[ \sigma_{\text{y}} = \frac{MC}{I} \quad M = F \times 1" = F \]

\[ = \frac{2670.65 (0.5)}{(0.75 \times 1')^2} = 21.37 \text{ ksi} \]

\[ \sigma_{\text{y}} = \sqrt{21.37^2 + 6.34^2} = 23.29 \text{ ksi} \]

\[ \text{FS.} = \frac{40}{23.29} = \frac{1.72}{1.72} \geq 1 \quad \text{OK} \]

based on yield
Fork

\[ F/2 = \frac{1043.2 \text{ lb}}{2} = 521.6 \text{ lb} \]

\[ A = \frac{1}{2} (1.75 - 0.75) = 0.0938 \text{ in}^2 \]

\[ \sigma = \frac{F/2}{A} = \frac{521.6}{0.0938} \]

\[ \sigma = 5.56 \text{ ksi} \]

\[ FS = \frac{40 \text{ ksi}}{5.56} = \frac{7.19}{1} > 3: \text{OK based on yield} \]

Seismic

\[ F = \frac{2670.65 \text{ lb}}{2} = 1335.3 \text{ lb} \]

\[ \sigma = \frac{1335.3}{0.0938} = 14.24 \text{ ksi} \]

\[ FS = \frac{40}{14.24} = 2.81 \geq 1.0: \text{OK} \]

Twist Threaded Rod

\[ \frac{1}{2} \text{ thread (Cartridge Brass)} \]

\[ E = 2.660 \text{ ksi for } \sigma_{max} = 265 \text{ ksi} \]
\[ A = \frac{\pi d^2}{4} = 0.0721 \]

\[ d = 0.303'' \text{ for } \varepsilon\text{-clip} \]

\[ \theta = \frac{3}{8}'' \]

\[ F = 1043.2 \text{ lb} \]

\[ S = \frac{F}{A} = \frac{1043.2}{0.0721} = 14.47 \]

\[ \text{FS} = \frac{55}{14.47} = 3.8 \]

\[ \sigma = \frac{S}{A} = \frac{1043.2}{0.023} = 45.36 \text{ ksi} \]

\[ \sigma = \frac{265}{45(116.1)} = 3.4 \text{ ksi} \]

\[ \text{Seismic: } \]

\[ F_s = 2670.65 \]

\[ S = \frac{F_s}{A} = \frac{2670.65}{0.0721} = 37.04 \text{ ksi} \]

\[ \text{FS} = \frac{55}{37.04} = 1.48 \]

\[ \text{Thread Stripping: } \]

\[ \frac{1}{2}'' \times 20 \text{ Brass } \]

\[ F = 1043.2 \text{ lb} \]

\[ L_e = 0.61'' \]

\[ A_s = \frac{3}{2} \left( D_n - \frac{1}{10} \right) L_e \]

\[ = \frac{3}{2} \left( \frac{1}{2} - \frac{1}{13} \right) (0.61) \]

\[ = 0.405 \text{ in}^2 \]
\[ \sigma = \frac{F}{A_s} = \frac{1043.2}{.405} \cdot 2.57 \text{ ksi} \]

\[ F.S. = \frac{55 \text{ ksi}}{\sqrt[3]{6(2.57)}} = 12.4 \geq 4 \text{ : OK} \]

Seismic:

\[ F = 2670.65 \]

\[ \sigma = \frac{F}{A_s} = 6.59 \text{ ksi} \]

\[ F.S. = \frac{55}{\sqrt[3]{6(6.59)}} = 4.82 \geq 1.25 \text{ : OK} \]

**Force to turn Crank:**

**NOTE:** The force (F = 1043.2 lb) used in the previous calculations was from the twist moment calculated when the assembly was at its extreme yaw angle. The mirror was off center 4 inches. These are rare cases, but the previous calculations needed to incorporate these assumptions just in case. Under normal circumstances, the mirror will be off centered by 1.5” (only because it is roughly centered by sight without measuring). Also at this time when the “twist” of the assembly would need to be adjusted, the yaw angle would be no more than 8 deg. For the nearest mirror sub-assembly (FABs). The calculation to find the force needed to turn the crank will be performed under the “normal circumstances” position!

Max angle for FABs:

\[ \Theta = \tan^{-1} \left( \frac{10.96}{.998} \right) \]

\[ \Theta = 6.3^\circ \]

To be conservative, use \( \Theta = 8^\circ \)
Find new twist moment:

\[ M_b = W_1 \left(1.52 + .99\right) + W_3 \left(2.9\right) \]

\[ = 276 \left(2.51\right) + 26.5 \left(2.9\right) = 700,4 \text{ in-lb} \]

\[ F = \frac{M_b}{2} = 350.2 \text{ lb} \]

\[ T = K(Dn)(F) \]

\[ K = 1 \text{ (see pg 27)} \]

\[ Dn = 0.5 \text{ in} \]

\[ T = 1.5(350.2) = 17.5 \text{ in-lb} \]

Force to turn wheel = \[ \frac{T}{\gamma W_{max}} \]

\[ = \frac{17.5}{2.3} = 7.6 \text{ lb} \]

Once again this is for the Fleury's mixer assembly. Most cases will be about \(\frac{4}{3}\) the force.
**Adapter Block**

- $W_1 = 27.6\text{ lb}$
- $W_3 = 26.5\text{ lb}$
- $W_6 = 6.7\text{ lb}$
- $W_6 = 17.9\text{ lb}$ (Shaft + Adapter block)

- $M_t = 2086.3\text{ (pg 21)}$

For $M_b$:

- $W_1(16.0) + W_3(1.92 + 7.15) + W_6(6.55) + W_4(2.7)$
- $= 4749.9$ in lb
\[ F_a = \frac{M_b}{2(4.118)} = \frac{4749.9}{2(4.113)} = 577.41 \text{ lb} \]

\[ F_s = \frac{\sum W}{3} \approx 109.03 \text{ lb} \]

Assumes only 3 out of the 6 screws take the load.

\[ F_s \text{ twist} = \frac{M_b}{3(2.375)} = \frac{2086.3}{3(2.375)} = 292.8 \text{ lb} \]

Assumes only 3 out of the 6 screws take the load.

\[ F_s = F_s \text{ twist} = 109.03 + 292.8 = 401.84 \text{ lb} \]

\[ \sigma = \frac{1}{A_b} \sqrt{F_a + 3F_s^2} \]

\[ A_b = 0.0567 \text{ in}^2 \text{ for M8} \]

\[ \sigma = \frac{1}{0.0567} \sqrt{(577.4)^2 + 3(401.84)^2} = 15.95 \text{ ksi} \]

\[ F_s = \frac{72.5}{15.95} = 4.5 \leq 4 \text{ : ok} \]

Pull out:

\[ F_a, \text{ worst case is when the rail plate has 0° rotation} \]

\[ M_b = W_1(18.05) + W_3(9.28) + W_5(1.55) + W_6(2.7) = 5319.1 \text{ inlb} \]
\[ Fa = \frac{Mb}{2(4.118)} = 646.6 \text{ lb} / \text{M6 bolt} \]

\[ A_b = \frac{\pi}{2} \left( D_n - \frac{D_b}{4} \right) L_c \]

\[ = \frac{\pi}{2} \left( 0.315 - 0.06 \right) \left( 0.63 \right) = 0.26 \text{ in}^2 \]

\[ \sigma = \frac{646.6}{0.26} = 2465.9 \text{ psi} = 2.466 \text{ ksi} \]

\[ FS = \frac{40 \times \sigma_y A_n}{\sqrt{3} \times 2466} = \sqrt{9.36} \geq 3 \text{ i.e. } \text{OK} \]

**Torsion:**

\[ T = K \cdot D_n \times F_p \]

\[ F_p \geq 1.6 Fa = 1034.6 \text{ lb} \]

\[ F_p \leq 0.65 \left( A_s \right) \left( S_n + \right) = 0.65 \left( 0.0567 \right) \left( 72\text{ ksi} \right) = 2672.0 \]

Choose \( F_p = 1850 \text{ lb} \)

\[ K = 0.3 \quad (SST \to AL, \text{ Dry}) \]

\[ T = 0.3 \left( \frac{8}{25.4} \right) \times 1850 = 174.8 \Rightarrow 175 \text{ in lb} \]

**Seismic:**

\[ SD_{x_1} = 172.2 \]

\[ SD_{x_2} = 16.5 \]

\[ SD_{x_3} = 4.2 \]

\[ SD_{y_1} = 430.6 \]

\[ SD_{y_2} = 41.3 \]

\[ SD_{y_3} = 3.1 \]

\[ SD_{y_4} = 10.5 \]

\[ SD_{y_5} = 27.9 \]

\[ \text{from page 32} \]
\[ M_b = 5341.3 \text{ lb (W_L does not add to twist, so it's the same as on pg 40)} \]

\[ M_b = (W_1 + SD_{xL})(16) + (W_3 + SD_{yL})(1.97 + 7.18) + (W_6 + SD_{yL})(6.55) + (W_6 + SD_{yL})(2.7) \]

\[ = 12,160 \text{ in. lb} \]

\[ F_a = F_{aSD} + F_{aMb} \]

\[ F_{aSD} = \frac{\sum SD_L}{b} = 34.02 \text{ lb} \]

\[ F_{aMb} = \frac{M_b}{3(4.13^2)} = 985.49 \text{ lb} \]

\[ F_a = 1019.5 \text{ lb} \]

\[ F_s = F_{s\text{twist}} + F_{s\text{weight}} \]

\[ F_{s\text{weight}} = \frac{M_b}{3(4.13^2)} = \frac{5341.3}{3(4.13^2)} = 749.66 \text{ lb} \]

\[ F_{s\text{weight}} = \frac{\sum SD_{yL} + \sum W}{3} = 279.13 \text{ lb} \]

\[ F_s = 1028.8 \text{ lb} \]

\[ G = \frac{1}{0.0567} \sqrt{(1019.5)^2 + 3(1028.8)^2} = 36.21 \text{ ksi} \]

\[ FS = \frac{72.5}{36.2} = \frac{2.0}{1.25} \geq 1.25 \text{ OK} \]

Pull out:
Find \( M_b \) with 300 yaw rotation:
\[ M_b = (W_t + SD_{y_b})(18.05) + (W_s + SD_{y_s})(9.25) + (W_6 + SD_{y_6})(6.55) \]
\[ + (W_b + SD_{y_b})(2.7) \]

\[ = 13,617.6 \]

\[ F_c = \frac{M_b}{2(4.118)} = 1655.4 \]

\[ \tau = \frac{1655.4}{1.26} = 6.37 \text{ ksi} \]

\[ FS = \frac{40}{\sqrt{3}(6.37)} = \frac{3.63}{1.16} \geq 1 : \text{ ok} \]

Robohand

Manufacturers rated load

\[ M_{b_{\text{max}}} = 6000 \text{ in lb} \]

\[ M_{b_{\text{max}}} = 9000 \text{ in lb} \]

Applied Load

\[ 5319.1 \text{ in lb} \]

\[ \uparrow \text{ non rotated position} \]

\[ 2086.3 \text{ in lb} \]

in both cases the applied load is under the maximum.